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HEATING DESIGN AND PRACTICE

Heating Design and Practice

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HEATING DESIGN AND PRACTICE

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II

For MARGARET HENDERSON

WHO HAS HELPED IN MANY UNDERTAKINGS

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PREFACE

This book has been prepared with two ends in view. First, it is intended to present to all persons concerned with heating, architects, engineers, teachers, students, and contractors—the basic theories of heating design with demonstrations of how these theories are worked out, followed in many instances by case histories and evaluations of results, taken from the practical experiences of the author.

These examples and evaluations are of importance because they point out how interdependent design and practice actually are. Admittedly a knowledge of design principles makes for better practice, but equally pertinent is a recognition of practice's capacity, on occasion or under special conditions, to show bluntly the need for a qualification of theory. In short, the confident prediction of satisfactory results must be predicated on a rounded knowledge of both design and practice.

For its second purpose, this book attempts to bring together working information on subjects and problems closely allied to heating which, sooner of later, are likely to demand the responsible attention of the heating engineer. In this category are those chapters which examine in detail the combustion of various fuels; the design of fuel handling and storage systems, including piping for gas and oil; the design of masonry chimneys and steel stacks; fireplaces, chimney pots, and incinerators; the characteristics of industrial refractories and insulation; the preparation of specifications and the analysis of bids.

Many of these data have been harvested from scattered sources, or directly from experience, and in consequence have not always been on hand in time of need. Workers and students in the heating field should therefore find in these chapters a most useful assistance.

Opportunity is taken here to acknowledge with gratitude the cooperation of the American Society of Heating and Ventilating Engineers, in permitting abstracts to be made of material originally appearing in the Guide. Without this cooperation the usefulness of this book would be seriously impaired.

Additional appreciation is expressed to Harry Unwin, to Heating and Ventilating, to the Byron Jackson Company, the Crane Company, the American Gas Association, the American Boiler Manufacturers Association, and to Babcock and Wilcox, all of whom made available to the author valuable data resulting from their researches and experience in their several fields.

Finally, each manufacturer who provided illustrations of his products is entitled to the appreciation of all who find value in this book. Without the cooperation of these concerns, the text would have been frequently drab and sometimes obscure.

Robert Henderson Emerick

Charleston, South Carolina
July, 1951

CONTENTS

Preface vii

1. Terms and Definitions 1
2. How Heating Loads Are Calculated 24
3. Designing a Steam Heating System 35
4. Heating Boilers and Steam Generators 61
5. Designing a Hot Water Heating System 97
6. Boiler Firing Equipment 127
7. Water Heaters and How to Size Them 156
8. Designing a Warm Air Heating System 180
9. Panels, Baseboards, and Unit Heaters 223
10. Fans and Filters 243
11. Space Heaters, Domestic and Industrial 253
12. District Steam Heating 260
13. The Heat Pump 276
14. Fuels and Fuel Handling 298
15. Combustion 341
16. Chimneys, Stacks and Flues 367
17. Heat Insulation and Refractories 396
18. Fireplace Design and Performance 412
19. How to Choose a Heating System 422
20. Incinerators and Destructors 431
21. Specifications and the Analysis of Bids 435

Index 443

CHAPTER 1

TERMS AND DEFINITIONS

The terms and definitions brought together in this chapter are working tools and as such become of prime importance.

After making our way through the list, we should undertake to rephrase the wording, bringing it within the limits of our individual understandings. When we can do this successfully without impairing the meaning of the term or definition, then we may regard the tool as our own and will be able to use it as the need appears.

Absolute Zero. This is the lowest temperature believed possible. Currently it is estimated at −459.8 degrees Fahrenheit and −273.2 degrees centigrade.

Absorbent. This is a substance that receives and makes part of itself another substance, the process being accompanied by a change in the physical or chemical characteristics of the receiver. Water readily functions as an absorbent and will receive anhydrous ammonia (water-free ammonia) to become aqua ammonia, thereby acquiring new characteristics, both physical and chemical.

Adsorbent. Here is a substance that receives within itself another substance, but does not change any of its own characteristics. Silica gel, for example, has a great affinity for moisture and will adsorb as much as 40 per cent of its own weight. However, the moisture resides within minute crevices and caves existing throughout the silica gel, and the host itself is unaffected. The action of a sponge in picking up water is similar; at these times the sponge becomes an adsorbent.

Adsorbents are important in air-conditioning processes, being used for the control of humidity.

Adiabatic. This term is commonly used in discussions and problems concerning the behavior of gases. The word is an adjective and describes a condition of the gas in which no heat can either enter the gas or leave it.

For example, when we refer to "adiabatic compression," we mean that the heat of compression is not removed and the temperature is allowed to rise. Air compressors operate close to the adiabatic despite the presence in some machines of water jackets for cooling. The heat is removed later, if necessary, by heat exchangers called "aftercoolers" or, if they are placed between the stages of compression, "intercoolers."

Aerosol. The word is a noun and is used to indicate, either collectively or individually, those motes of matter, both solid and liquid, which float around in the atmosphere. Fog, smoke, and dust are formed of aerosol and are aerosol.

Aspect Ratio. This term designates the relationship of the length of a grille to the width of the grille. Thus, a grille 20 inches long by 10 inches wide has an aspect ratio of 2.

The aspect ratio affects the shape of the air jet emerging from the grille, but precisely what is the best ratio to choose is largely a matter of opinion. According to reports by the American Society of Heating and Ventilating Engineers, air jets from a rectangular grille having a low aspect ratio tend to become cone-shaped very close to the outlet. From this we might assume that results will be reasonably uniform for any aspect ratio, provided that it is small. We can therefore size our grilles with considerable regard for appearance and the limitations of location.

Blow. This is the distance an air stream travels on emerging from a grille or register, before its velocity falls to 50 feet per minute, when this velocity is measured along the axis of the stream. Sometimes it is called "throw."

Boiler Horsepower. One boiler horsepower is developed by the evaporation of 34.5 pounds of water at 212 degrees Fahrenheit to steam at 212 degrees Fahrenheit. This is equal to a heat input of 34.5 × 970.3 Btu per pound, or 33,475 Btu per hour.

Since steam boilers invariably operate at steam and water temperatures quite different from 212 degrees Fahrenheit, we must determine the horsepower of a given boiler for the conditions under which it is operating by comparing its rate of evaporation with the standard.

For example, suppose that we have a heating boiler that is producing steam at 14 pounds per square inch gauge pressure from feed water having a temperature of 140 degrees Fahrenheit. Referring to Table 1.2, which contains the properties of steam, we observe that the total heat in each pound of this steam is 1163.3 Btu, approximately, by interpolation.

Of this total heat, 140 F − 32 F, or 108 Btu, already is present in the feed water; consequently, the boiler is called upon to provide 1163.3 − 108, or 1055.3 Btu, for each pound of steam.

Since standard conditions for horsepower stipulate only 970.3 Btu per pound to be added by the boiler, in the case assumed the boiler must invest 1055.3/970.3, or 1.09 Btu, for each Btu contemplated under the standard. In practical applications, this means that a boiler rated at 50 horsepower by the manufacturer will deliver only 50/1.09, or approxi-

TABLE 1.1. DRY SATURATED STEAM: PRESSURE TABLE*

Abs. pres., psi p	Temp., F t	Specific volume		Enthalpy		
		Sat. liquid v_f	Sat. vapor v_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4
10	193.21	0.01659	38.42	161.17	982.1	1143.3
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4
15	213.03	0.01672	26.29	181.11	969.7	1150.8
20	227.96	0.01683	20.089	196.16	960.1	1156.3
25	240.07	0.01692	16.303	208.42	952.1	1160.6
30	250.33	0.01701	13.746	218.82	945.3	1164.1
35	259.28	0.01708	11.898	227.91	939.2	1167.1
40	267.25	0.01715	10.498	236.03	933.7	1169.7
45	274.44	0.01721	9.401	243.36	928.6	1172.0
50	281.01	0.01727	8.515	250.09	924.0	1174.1
55	287.07	0.01732	7.787	256.30	919.6	1175.9
60	292.71	0.01738	7.175	262.09	915.5	1177.6
65	297.97	0.01743	6.655	267.50	911.6	1179.1
70	302.92	0.01748	6.206	272.61	907.9	1180.6
75	307.60	0.01753	5.816	277.43	904.5	1181.9
80	312.03	0.01757	5.472	282.02	901.1	1183.1
85	316.25	0.01761	5.168	286.39	897.8	1184.2
90	320.27	0.01766	4.896	290.56	894.7	1185.3
95	324.12	0.01770	4.652	294.56	891.7	1186.2
100	327.81	0.01774	4.432	298.40	888.8	1187.2
110	334.77	0.01782	4.049	305.66	883.2	1188.9
120	341.25	0.01789	3.728	312.44	877.9	1190.4
130	347.32	0.01796	3.455	318.81	872.9	1191.7
140	353.02	0.01802	3.220	324.82	868.2	1193.0
150	358.42	0.01809	3.015	330.51	863.6	1194.1
160	363.53	0.01815	2.834	335.93	859.2	1195.1
170	368.41	0.01822	2.675	341.09	854.9	1196.0
180	373.06	0.01827	2.532	346.03	850.8	1196.9
190	377.51	0.01833	2.404	350.79	846.8	1197.6
200	381.79	0.01839	2.288	355.36	843.0	1198.4
250	400.95	0.01865	1.8438	376.00	825.1	1201.1
300	417.33	0.01890	1.5433	393.84	809.0	1202.8
350	431.72	0.01913	1.3260	409.69	794.2	1203.9
400	444.59	0.0193	1.1613	424.0	780.5	1204.5
450	456.28	0.0195	1.0320	437.2	767.4	1204.6
500	467.01	0.0197	0.9278	449.4	755.0	1204.4
550	476.94	0.0199	0.8424	460.8	743.1	1203.9
600	486.21	0.0201	0.7698	471.6	731.6	1203.2
650	494.90	0.0203	0.7083	481.8	720.5	1202.3
700	503.10	0.0205	0.6554	491.5	709.7	1201.2
750	510.86	0.0207	0.6092	500.8	699.2	1200.0
800	518.23	0.0209	0.5687	509.7	688.9	1198.6
850	525.26	0.0210	0.5327	518.3	678.8	1197.1
900	531.98	0.0212	0.5006	526.6	668.8	1195.4
950	538.43	0.0214	0.4717	534.6	659.1	1193.7
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4
1300	577.46	0.0227	0.3293	585.4	593.3	1178.6
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3
3206.2	705.40	0.0503	0.0503	902.7	0	902.7

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* Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York. Copyright, 1937, by Joseph H. Keenan and Frederick G. Keyes.

TABLE 1.2. PROPERTIES OF SUPERHEATED STEAM*

Abs. press., psi (Sat. temp.)		Temperature, F								
		200	300	400	500	600	700	800	900	1000
	v....	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5
1	h...	1150.4	1195.8	1241.7	1288.3	1335.7	1383.7	1432.8	1482.7	1533.5
(101.74)	s....	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283
	v....	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87
5	h...	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4
(162.24)	s....	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509
	v....	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92
10	h...	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2
(193.21)	s....	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744
	v....	......	30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13
14.696	h...	......	1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1
(212.00)	s....	......	1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	1.0958	2.1319
	v....	......	22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44
20	h...	......	1191.6	1239.2	1286.2	1334.4	1382.9	1432.1	1482.1	1533.0
(227.96)	s....	......	1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978
	v....	......	11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70
40	h...	......	1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4
(267.25)	s....	......	1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212
	v....	......	7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454
60	h...	......	1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9
(292.71)	s....	......	1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762
	v....	......	......	6.220	7.020	7.797	8.562	9.322	10.077	10.830
80	h...	......	......	1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3
(312.03)	s....	......	......	1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442
	v....	......	......	4.937	5.589	6.218	6.835	7.446	8.052	8.656
100	h...	......	......	1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8
(327.81)	s....	......	......	1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193
	v....	......	......	4.081	4.636	5.165	5.683	6.195	6.702	7.207
120	h...	......	......	1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2
(341.25)	s....	......	......	1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990
	v....	......	......	3.468	3.954	4.413	4.861	5.301	5.738	6.172
140	h...	......	......	1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7
(353.02)	s....	......	......	1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817
	v....	......	......	3.008	3.443	3.849	4.244	4.631	5.015	5.396
160	h...	......	......	1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1
(363.53)	s....	......	......	1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667
	v....	......	......	2.649	3.044	3.411	3.764	4.110	4.452	4.792
180	h...	......	......	1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6
(373.06)	s....	......	......	1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534
	v....	......	......	2.361	2.726	3.060	3.380	3.693	4.002	4.309
200	h...	......	......	1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0
(381.79)	s....	......	......	1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415
	v....	......	......	2.125	2.465	2.772	3.066	3.352	3.634	3.913
220	h...	......	......	1206.5	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5
(389.86)	s....	......	......	1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308
	v....	......	......	1.9276	2.247	2.533	2.804	3.068	3.327	3.584
240	h...	......	......	1202.5	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9
(397.37)	s....	......	......	1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209
	v....	......	......	......	2.063	2.330	2.582	2.827	3.067	3.305
260	h...	......	......	......	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3
(404.42)	s....	......	......	......	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118
	v....	......	......	......	1.9047	2.156	2.392	2.621	2.845	3.066
280	h...	......	......	......	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8
(411.05)	s....	......	......	......	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033
	v....	......	......	......	1.7675	2.005	2.227	2.442	2.652	2.859
300	h...	......	......	......	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2
(417.33)	s....	......	......	......	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954
	v....	......	......	......	1.4923	1.7036	1.8980	2.084	2.266	2.445
350	h...	......	......	......	1251.5	1310.9	1365.5	1418.5	1471.1	1523.8
(431.72)	s....	......	......	......	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777
	v....	......	......	......	1.2851	1.4770	1.6508	1.8161	1.9767	2.134
400	h...	......	......	......	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4
(444.59)	s....	......	......	......	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623

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* Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York. Copyright, 1937, by Joseph H. Keenan and Frederick G. Keyes.

mately 46 horsepower, under the conditions of 14 pounds per square inch gauge pressure and 140-degree Fahrenheit feed water.

A boiler horsepower is sometimes designated in terms of heating surface, 1 horsepower being considered the equal of 10 square feet. Although this view enables us to compare boilers on the basis of heating surface being provided, it is no measure whatever of the steaming capacity of the boiler. We should buy boilers on the basis of pounds per hour output under the conditions of pressure and feed-water temperatures that will exist and under which the boilers must operate.

Btu. This combination of letters stands for "British thermal unit." It is a measure of heat, and for all practical purposes we can consider one Btu to represent the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit.

Conductance. Conductance is of two types: surface and thermal.

Surface conductance refers to the movement of heat from any given surface to contacting air or other fluid, and vice versa. It is expressed in terms of Btu per hour per square foot of surface per degree of temperature difference between the two contacting materials.

Thermal conductance refers to the movement of heat *through* a material. It is expressed in Btu per hour per square foot per degree of temperature difference for any given shape and size of material. Thus, if we have a brick wall, we can say it has a conductance of such and such a value, while a stone wall would have a conductance of another value, even though the thickness of the wall remained unchanged. In other words, conductance is a function of the material for any selected thickness.

Conductivity. This word applies to the conductance of a material expressed in Btu per hour per square foot per degree of temperature difference *per inch of thickness.* In the heating field, conductivity is meant when we refer to the k factor of a material. For example, rock wool, an insulating material, has a conductivity (k factor) of 0.27 Btu per hour per square foot per degree of temperature difference per inch of thickness.

Convection. Convection designates the movement that occurs in any fluid, air or liquid, as the result of differences in density acted on by the force of gravity. Thus, if we apply heat to the bottom of a water tank, the hot water so formed soon moves to the top of the tank since the heat decreases its density, and the cold water is enabled therefore, by gravity, to push in beneath it.

If heat is applied steadily and continuously, we soon establish a definite circulation inside the tank, with the warmed water moving upward and the cold water descending. This movement of the water is called "natural convection."

Sometimes this movement is expedited by a pump, and then we call it "forced convection."

We can use this reaction to our advantage in times of inordinately cold weather by directing the blow of a fan across a radiator. The heat output of the radiator is increased substantially by this recourse.

Convectors. Convectors differ fundamentally from radiators by being designed to foster air movement across their heated surfaces. To achieve this objective, the hot coil is enclosed in a metal cabinet, with openings at the bottom and top, to simulate a modified chimney. In this way, some measure of concentrated air movement across the coil is obtained, usually by natural convection.

Convectors may be recessed into the wall, partly recessed, free-standing, or suspended from the wall.

If desired, convectors can be purchased which incorporate a fan inside the housing, thus greatly augmenting the heat output.

Decibel. This word is the measure of sound. We hear sounds as the manifestation of vibrations and loudness, or intensity.

The human ear recognizes vibrations over a certain range and normally makes no response to either very high frequency or very low frequency. We all are familiar, for example, with the dog whistle that is soundless to human perception, but is instantly identified by the animal. The human range extends from 20 to 20,000 cycles per second.

In order to create a scale of sound, the vibrations are used to generate a minute electric current. At the threshold of human hearing this current amounts to a tiny percentage of a watt and is expressed as 10^{-16} watt per square centimeter, at a frequency of 1,000 cycles per second.

By comparing the current at any observed condition of sound with this standard, we can arrive at a definite value for sound. Instruments are available commercially that read directly in decibels, a microphone being used to pick up the sound, and a vacuum-tube circuit serving to amplify it.

Although we use the term "decibel" as a matter of convenience, the basic unit is the "bel," being in value 10 decibels. The name was chosen to honor Alexander Graham Bell.

The average sound level observed will differ somewhat for any selected place, depending on the observer and the technique employed with the microphone. However, simply as an illustration of what might be expected, the ASHVE Guide suggests an average of 14 decibels for radio broadcasting studios, 40 decibels for homes, 30 decibels for churches, and 60 decibels for general offices.

Reports from a fan manufacturer indicate that sound levels in business concerns are rather higher. With business machines in operation, the range varies from 95 to 103 decibels; in a general office, from 80 to 90; for a restaurant, from 60 to 85 decibels.

The fact is, however, that in practice, if we are called on to reduce an existing noise level, we must do so regardless of the decibel reading.

Degree-day. Here is a unit that enables us to compare the heating requirements of one month with another, and of one city with another. By calculating fuel consumption per degree-day for any given period of time, we can arrive at an estimate on the efficiency of our combustion.

Degree-day calculations start at a temperature of 65 degrees Fahrenheit, as recorded by the local weather bureau. This temperature is chosen as the starting point on the premise that all lower temperatures will entail heating, whereas higher temperatures need none.

If the mean temperature is 30 degrees Fahrenheit, for example, the difference between this mean and 65 degrees Fahrenheit is 35 degrees for the day involved, and for this one day we have accumulated 35 degree-days. Thus, by definition, the degree-days for any one day is the difference between the mean temperature of that day and 65 degrees Fahrenheit.

The mean temperature is taken as the mean between the highest and lowest temperatures recorded throughout a 24-hour period. If the thermometer shows zero at midnight and 40 degrees Fahrenheit in the afternoon, and these are the lowest and highest temperatures recorded during the 24 hours, then the mean temperature for that day is $0 + {}^{40}\!/_{2} = 20$ F, and the degree-days accumulated will be $65 - 20 = 45$ degree-days.

Table 1.3 presents the average degree-days for cities in the United States and Canada.

Dew Point. The amount of moisture in the form of water vapor that we find present in air is limited ultimately by the temperature of the air. Thus, for any given temperature, a cubic foot of air will accommodate a fixed number of grains, and not one grain more. (Incidentally, there are 7,000 grains in a pound.)

In this condition, the air is said to be "saturated." If we raise the temperature of saturated air, it ceases to be saturated at the higher temperature and can accept additional water vapor. Conversely, if we lower the temperature of saturated air, some of the moisture is squeezed out, the amount thus precipitated being governed by the extent to which we lower the temperature.

The temperature at which precipitation takes place is called the "dew point." On the psychrometric chart (Fig1.1), the dew point for various dry-bulb temperatures is shown as a curved line, this line sometimes being referred to as the "saturation line," since it represents a condition of 100 per cent relative humidity.

Appreciation of the dew point is of great importance in heating and air-conditioning activities. For illustration, if we have designed a panel heating system and attempt to secure a measure of cooling by pumping cool water through the coils in summer, we must be careful to maintain water temperatures above the existing dew point, or the walls and ceiling surfaces over the coil will sweat.

Table 1.3. Annual Degree-days of Various Cities Indicative of Areas

City	*Degree-days*	*City*	*Degree-days*
Eastport, Me	8,520	Edmonton, Canada	10,285
Boston, Mass	6,045	Vancouver, Canada	5,573
New York, N.Y	5,274	Winnepeg, Canada	10,980
Philadelphia, Pa	4,737	Boise, Idaho	5,552
Washington, D.C	4,626	Billings, Mont	7,213
Norfolk, Va	3,350	Bismarck, N.D	9,192
Charleston, S.C	1,769	Minneapolis, Minn	7,850
Miami, Fla	185	Chicago, Ill	6,077
New Orleans, La	1,024	Cleveland, Ohio	6,155
Houston, Tex	1,157	St. Louis, Mo	4,585
El Paso, Tex	2,428	Louisville, Ky	4,180
Albuquerque, N.M	4,298	Memphis, Tenn	2,950
Santa Fe, N.M	6,123	Denver, Colo	5,874
Phoenix, Ariz	1,405	Salt Lake City, Utah	5,555
Flagstaff, Ariz	7,241	Cheyenne, Wyo	7,466
Los Angeles, Calif	1,504	Reno, Nev	5,892
San Francisco, Calif	3,264	Montreal, Canada	8,400
Seattle, Wash	4,934	Toronto, Canada	7,374
Nome, Alaska	14,580	Halifax, Canada	7,614

A similar problem arises during the winter as the result of more or less humid air striking cold walls and windows; consequently, humidification of inside air should be controlled with caution. If a space is heated by a nonvented gas burner (and the character of the gas is immaterial), the process of combustion produces considerable moisture in the air, since the hydrogen in the gas burns to water vapor. The amount of water vapor so created is discussed in Chap. 15, Combustion, and we can readily calculate the resulting humidity for any particular installation.

Density. By definition, density is the weight per unit of volume.

In the English system, we express density as pounds per cubic foot. In heating calculations, we are much concerned with the relationships of density and volume, since we must know both of these for air, steam, and gases in order to design ducts and piping, stacks and flues.

If we ascertain the density of any substance for a given temperature, we readily can convert that knowledge into volume of the substance per pound by solving the fraction, $\frac{1}{\text{pounds per cubic foot}}$. For example, dry

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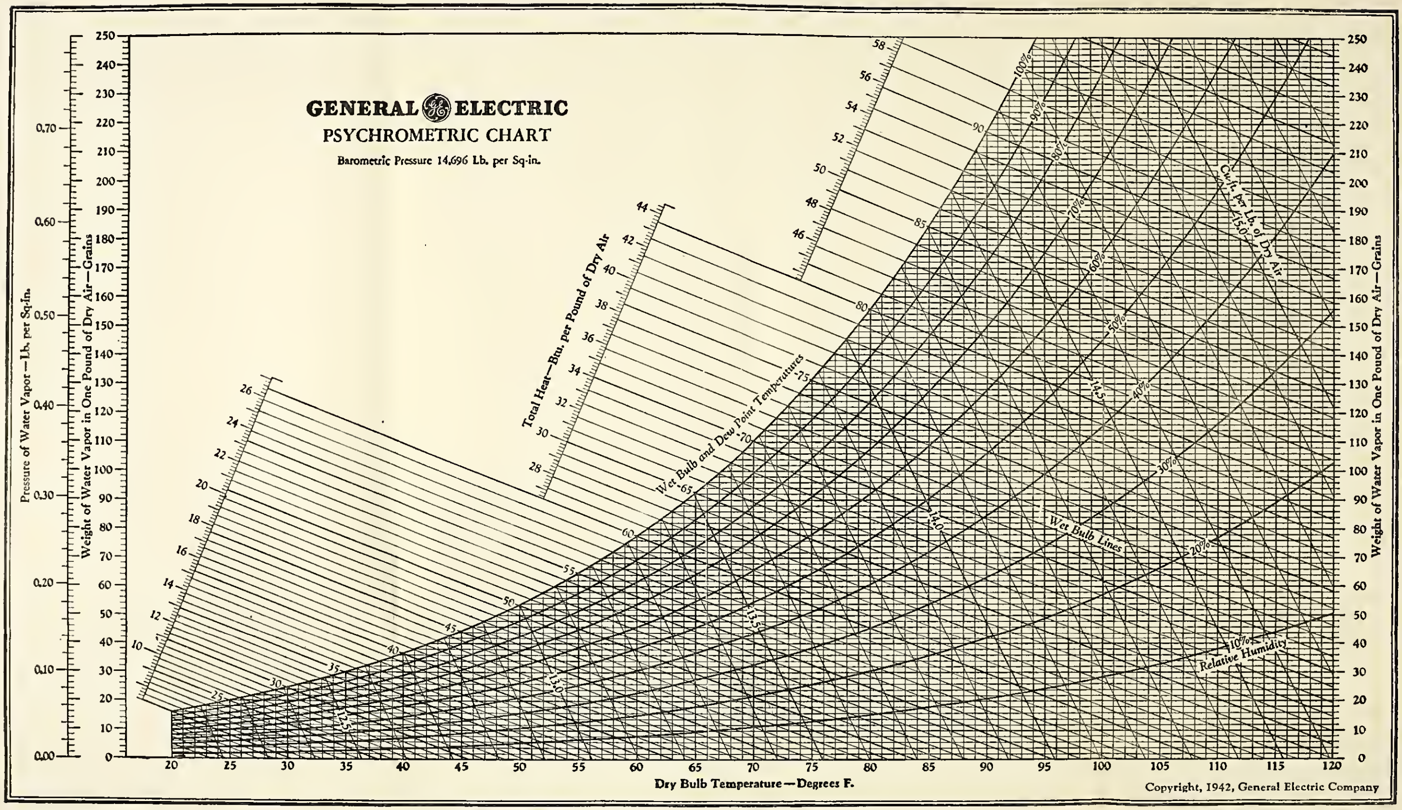

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EXPLANATION

Fig. 1

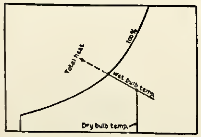


Fig. 2

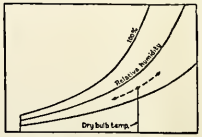


Fig. 3

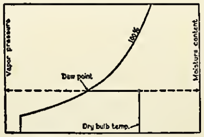
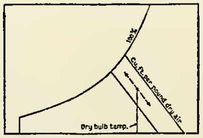


Fig. 4



If two properties of air are known, all properties may be found as follows:

DRY-BULB TEMPERATURE is read directly by following vertically down to the bottom scale.

WET-BULB TEMPERATURE (Fig. 1) is read directly at the intersection of the wet-bulb line with the 100 per cent relative humidity line (saturation curve). The scale is marked along the 100 per cent line.

RELATIVE HUMIDITY (Fig. 2) is read directly from the curved lines marked Relative Humidity. For a point between the lines, estimate by distance.

MOISTURE CONTENT, or Absolute Humidity (Fig. 3), is read directly from horizontal lines with scales to the right and left of the chart, and is the weight of water vapor contained in a quantity of air and water-vapor mixture which would weigh 1 lb if all water vapor were extracted.

DEW-POINT TEMPERATURE (Fig. 3) is read at the intersection of a horizontal line of given moisture content with the 100 per cent relative humidity line.

TOTAL HEAT (Fig. 1) is read directly by following the wet-bulb line to the scale marked Total Heat. Total Heat refers to a quantity of air and water-vapor mixture which would weigh 1 lb if all water vapor were extracted, and includes the heat of the water vapor.

SPECIFIC VOLUME (Fig. 4) is read directly from the lines marked Cu ft per Lb of Dry Air. For points between lines, estimate by distance. Specific volume is the volume occupied by a quantity of air and water-vapor mixture which would weigh 1 lb if all water vapor were extracted.

VAPOR PRESSURE (Fig. 3) corresponding to a given moisture content is read directly from the left-hand scale marked Pressure of Water Vapor.

Fig. 1.1. Psychrometric chart.

EXAMPLES

A. Air at 80 deg dry bulb and 65 deg wet bulb.

Relative Humidity	45 per cent
Moisture Content	68.5 gr per lb
Dew Point	56.8 F
Total Heat (includes heat of 1 lb of dry air and heat of 68.5 grains of water vapor)	30.0 Btu per lb
Specific Volume	13.81 cu ft per lb
(Actual weight of 13.81 cu ft. = 1.0098 lb; 1 lb for the dry air and 68.5 grains for the moisture content, 7000 grains = 1 lb)	
Vapor Pressure	0.228 lb per sq in.

B. Air at 75 deg wet bulb and 67 deg dew point.

Dry-bulb Temperature	94.7 F
Relative Humidity	40.0 per cent
Moisture Content	99.2 gr per lb
Total Heat	38.5 Btu per lb
Specific Volume	14.28 cu ft per lb
Vapor Pressure	0.328 lb per sq in.

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pound by solving the fraction, pounds per cubic foot air at 70 degrees Fahrenheit weighs 0.0749 pound per cubic foot, and the corresponding volume is 1/0.0749 = 13.348 cubic feet per pound. The conversion of volume to density is equally simple, the procedure being

$$\frac{1}{\text{cubic feet per pound}}.$$

Enthalpy. This term is used to express heat content, or total heat. Thus, from Table 1.1, we observe that the enthalpy (or total heat) of steam at 100 pounds per square inch absolute pressure, saturated, is 1187.2 Btu per pound.

Equivalent Direct Radiation. Under standard conditions of 215-degree Fahrenheit steam and 70-degree Fahrenheit ambient air, a steam radiator emits 240 Btu per hour per square foot of radiator surface.

Since the occurrence of such standard conditions in everyday practice is purely coincidental, we must refer the actual conditions, whatever they happen to be, to this standard of 240 Btu per hour per square foot of surface in order to size our boiler. Thus, if we have calculated a heat load for any given building to be 240,000 Btu per hour, the boiler will be called upon to meet a connected demand of 240,000/240 or 1,000 square feet of "equivalent direct radiation."

Boiler manufacturers frequently catalogue the capacities of their products in square feet of equivalent direct radiation when these boilers are intended for low-pressure heating service. The big industrial boilers generating steam at high pressure, from which perhaps steam is abstracted for incidental heating, are sized differently, usually in pounds of steam output per hour under the desired conditions, or else in boiler horsepower.

Radiator, convector, and unit heater capacities are taken directly from the manufacturers' catalogues, since we have considerable choice of heights, lengths, and depths to meet our particular conditions.

The term "equivalent direct radiation" is a very useful one in that it enables us to appreciate instantly the magnitude of a heating demand. It is frequently written simply EDR.

Equivalents. The interchangeability of various units of measurement is of essential importance to all engineering design. For example, the mechanical horsepower of an electric motor is charged for on the electric bill in kilowatthours; consequently, in computing operating costs for a design we must multiply each horsepower by 0.746 to ascertain the equivalent number of kilowatts.

Table 1.4 makes available these useful conversion equations.

Free Air. The expression "free air" is in common use by fan manufacturers to designate the capacity of their equipment.

The term has two meanings, and we should be careful that the meaning which is applicable to our particular conditions is the one by which we select a fan for size.

The standard meaning of free air is air at sea-level elevation and at a temperature of 60 degrees Fahrenheit.

TABLE 1.4. USEFUL EQUIVALENTS

Item	*Equivalent to*
1 British thermal unit (Btu)	778.3 foot-pounds 0.293 watthours 252.0 calories
1 boiler horsepower (boiler hp)	33,475 Btu per hour 9.809 kilowatts
1 horsepower (hp), mechanical	33,000 foot-pounds per minute 550 foot-pounds per second 42.4 Btu per minute 0.746 kilowatts
1 kilowatthour (kwhr)	3413 Btu
1 kilowatt (kw)	1.341 horsepower
1 ton of refrigeration	288,000 Btu per 24 hours 12,000 Btu per hour
1 U.S. gallon (U.S. gal)	0.1337 cubic feet
1 cubic foot (cu ft)	7.481 U.S. gallons
1 bushel (bu)	1.244 cubic feet
1 pound avoirdupois (lb av.)	7,000 grains
1 pound per square inch pressure (psi)	144 pounds per square foot 2.042 inches of mercury 2.31 feet of water 27.71 inches of water
1 inch water gauge (in. W.G.)	0.036 pounds per square inch 0.577 ounces per square inch 5.197 pounds per square foot
1 inch of mercury (in. Hg)	0.433 pounds per square inch 13.57 inches of water

The other meaning concerns the conditions of elevation and temperature at the site of the fan operation.

Obviously, as elevations increase and as temperatures rise above the standard 60 degrees Fahrenheit, the volume of the air to be handled expands. Thus, under the latter conditions, the fan actually delivers less air *by weight*, and this reduced output may fall short of our needs.

Freon. This proprietary refrigerant is being discussed here because a great majority of heat-pump circuits are operating with Freon as the refrigerant.

Freon is the name given to a whole family of refrigerants originated and manufactured by the Kinetic Chemicals Company. Chemically, these refrigerants are chlorinated hydrocarbons.

The Freon most commonly used is known as F-12 commercially, its chemical name being dichlorodifluoromethane. The other members of the family are F-11, F-113, F-114, F-21, F-22.

Table 13.1 presents the properties of F-12. For calculations concerning other members of the family, a table of properties should be obtained direct from the manufacturer.

Head. The term "head" refers to a pressure, expressed either in feet of water, inches of water, or pounds per square inch. Feet and inches of course are readily convertible to pounds per square inch.

There are several kinds of head. First, we have the *static head* caused by differences in elevation. For example, if we have a condensate pump in a pit which discharges into a boiler at a higher elevation, the static head equals the distance in feet between the center of the pump inlet and the center of the discharge at the boiler connection.

The next head to consider is the *friction head.* As the water flows through piping, changes direction through tees and elbows, and threads its way through valves, it meets considerable resistance to its passage as the result of friction. The amount of this friction, usually expressed in equivalent feet of straight pipe for the fittings, has been determined (though not always identically) by various investigators.

There are also the *entry head* and *velocity head*, but both of these are of relative unimportance in heating work, and generally may be neglected. The entry head is really a form of friction head, and refers to the resistance to flow encountered by the fluid when entering a pipe. The velocity head represents the height in feet through which a body would fall in order to equal the velocity of the fluid in feet per second. We can calculate the velocity head from the formula $v = \sqrt{2gh}$. In this formula, the g is the gravity factor and is 64.32, the velocity of the flow is whatever it happens to be, and the h, representing the velocity head, is the unknown.

The *total discharge head* is built up from these various component heads, and it represents the ultimate load on the pump. Normally, we compute the total discharge head by adding together the static head, the friction head, and the pressure needed at the point of discharge. Occasionally, we may encounter a situation in which there exists an actual pressure on the suction side of the pump, instead of the usual suction lift. In this case we subtract the value of the suction pressure, either in feet or pounds per square inch, from the summation of the other heads on the discharge side of the pump, and the total discharge head is thereby obtained.

Dynamic head is sometimes used in lieu of total discharge head, but the meaning is identical.

We should note that the work a pump performs is mostly increment work, that is, it adds pressure to whatever pressure may exist on the suction side. This characteristic is used most conveniently at times by hooking up two pumps of low individual heads in series and thereby obtaining a high head discharge.

Humidity. This word comes from the Latin *humidus,* and means moist. In heating work, we are concerned with moisture in the form of water vapor, and as such, we are further obliged to consider it from two viewpoints.

The first of these viewpoints we identify as relative humidity. For saturated atmospheres, the relative humidity is 100 per cent. For conditions less than saturated, the water vapor present is expressed as a percentage of the saturated condition. Thus, 80 per cent relative humidity is also 80 per cent saturated.

Since commercial instruments, the psychrometer and the hygrometer, measure the moisture present in terms of relative humidity, an understanding of the term is important. Incidentally, the prefix *hygro-* is the Greek for wet.

With the relative humidity ascertained, we can use a psychrometric chart and determine the *absolute humidity,* which refers to the number of *grains* of moisture present in the air.

Knowledge of absolute humidity is essential in problems of dehumidification, since we can do little if the amount of moisture to be removed is not known.

Inches Water Gauge. When we are dealing with small pressures, we find it convenient to express these pressures in inches of water, rather than in pounds per square inch.

The water gauge is usually observed in the form of a manometer, or U tube, and as such functions commonly as an indicator of draft in boilers, of pressure and velocity of air and gas movements in ducts, and of similar small forces.

The standard temperature for water in a water gauge is 62 degrees Fahrenheit, and at this temperature, an inch of water pressure is equivalent to 0.0361 pound per square inch (psi). This value is determined by the simple fraction 62.355/1,728, which is the weight of a cubic foot of water divided by the number of cubic inches in a cubic foot.

Inches of Mercury. Mercury column gauges are used to measure subatmospheric pressures. Approximately 30 inches of mercury (in. Hg) will equal the atmospheric pressure at sea level of 14.7 pounds per square inch. For quick figuring, some engineers consider a half inch of mercury as being equal to a pressure of ¼ pound per square inch. Just to create perspective, we might note that 7 inches of water are required, approximately, to produce the same ¼ pound pressure.

Isothermal. This term has a significance exactly opposite to adiabatic. If we could compress air or a gas isothermally, we would remove the heat of compression as rapidly as it was created.

By definition, isothermal is an adjective meaning constant temperature.

Latent Heat. Latent heat is insensible heat, which means that it cannot be measured on a thermometer.

It is the heat added to water which changes the water into steam, without affecting the temperature. Thus, if we have a pound of water under atmospheric pressure and at a temperature of 212 degrees Fahrenheit, we cannot change the water into steam until we add 970.3 British thermal units (Btu).

These 970.3 Btu are termed the latent heat of evaporation.

Farther down the thermometer latent heat appears again, this time being the heat that converts ice into water. Here it is called the heat of fusion; sometimes the latent heat of freezing.

In quantity of heat involved, the heat of fusion is much less than the heat of evaporation, only 144 Btu being required to change a pound of ice into a pound of water.

The actual amount of heat required for 1 pound of evaporation, depends on the pressure at which the evaporation takes place. For example, at 100 pounds per square inch gauge (psig), approximately 880.6 Btu will be sufficient, at 300 psig, about 804.5 will do, and at 3,221 psig, the critical pressure, none whatever is required.

Logarithmic Mean Temperature Difference. Since the transfer of heat from one material to another depends on their difference in temperature, we must understand how this difference is calculated.

As an illustration, suppose that we are interested in a fuel-oil preheater which will warm the oil from 70 to 170 F, when supplied with steam at 250 F. The entering temperature difference is 250 − 70, or 180 degrees. The leaving temperature difference is 250 − 170, or 80 degrees.

The arithmetic mean temperature difference obviously is

$$(180 + 80)/2 = 130 \text{ degrees}$$

Unfortunately, when the larger temperature difference is more than twice the smaller temperature difference, the arithmetic mean temperature difference is too inaccurate for use; consequently, we must compute the logarithmic mean temperature difference in this case. The equation then becomes

Logarithmic mean temperature difference

$$= \frac{180 - 80}{\log_e {}^{180}\!/_{80}}$$

$$= \frac{100}{0.8109} = 123.3 \text{ F} \qquad \text{approximately}$$

Log_e is the Naperian, hyperbolic, or natural logarithm. If we do not have a table of Naperian logarithms handy, we can use the common logarithms, multiplying by 2.30.

Mol. If we evaluate the weight of a substance in terms of its molecular weight, then we can deal in mols. A mol of the substance consists of as many pounds as may be required to equal its molecular weight. Thus a mol of hydrogen equals 2 pounds of hydrogen, a mol of oxygen, 32 pounds of oxygen, a mol of air, approximately 29 pounds of air.

The mol concept can be used advantageously in the solving of gas problems, since it provides us with a quick insight into gas densities. For instance we can determine the volume of any gas at any pressure and any temperature by exercising the following equation:

$$v = \frac{1{,}544 \times T}{P}$$

where v = volume of 1 mol of the gas
T = absolute temperature, F
P = absolute pressure, pounds per square foot (if pressure happens to be known in pounds per square inch, which is our usual form of expression, then we must convert it to pounds per square foot, multiplying by 144)

If we consider the case of atmospheric pressure and 32 F, and insert these values in the equation, the solution indicates that the volume of 1 mol of a gas, any gas, even air, is approximately 359 cubic feet. Now if the gas is oxygen, which has a molecular weight of 32, then 359 cubic feet of oxygen weighs 32 pounds, but if the gas should be hydrogen, with a molecular weight of 2, then the 359 cubic feet weighs only 2 pounds. By changing the T and P values of the equation, we can determine the

TABLE 1.5. MOLECULAR WEIGHTS OF VARIOUS SUBSTANCES

Substance	Symbol	Molecular weight
Acetylene	C_2H_2	26.03
Air	......	29.00
Ammonia	NH_3	17.03
Chlorine	Cl	70.92
Carbon dioxide	CO_2	44.00
Carbon monoxide	CO	28.00
Ethylene	C_2H_4	28.04
Helium	He	4.00
Hydrogen	H	2.02
Methane	CH_6	16.04
Methyl chloride	CH_3Cl	50.49
Nitrogen	N	28.02
Oxygen	O	32.00
Sulphur dioxide	SO_2	64.06

density and volume of any gas under any conditions we happen to encounter.

Many combustion engineers use the mol as a fast and satisfactory tool in dealing with the changing conditions of flue gases. Incidentally, the word is pronounced "mole."

Table 1.5 presents some commonly encountered and useful molecular weights.

Reynolds Number. The flow of any fluid, whether it be gas or liquid, is affected by several factors occurring along its course, particularly the friction of the surfaces with which it is in contact and its own internal resistance to flowing, called "viscosity."

The various formulas for fluid flow do not always take viscosity into consideration; consequently, the results obtained by solving the same problem with different formulas will vary also. For instance, the Hazen and Williams formula, long more or less standard in water computations, does not consider viscosity. The Fanning formula on the other hand, which is written

$$h = \frac{fLv^2}{2gD} \qquad \text{for liquids}$$

and

$$P = \frac{fLv^2p}{24 \times gd} \qquad \text{for gas}$$

includes f as a friction factor, which is determined by means of the Reynolds number.

The Reynolds number represents a drawing together into one computed figure of all the factors which affect the friction loss incurred by a flowing fluid. The number is reached by solving the equation:

$$R = \frac{Dvp}{u}$$

where R = Reynolds number
D = diameter of the pipe, ft, in., or cm (centimeters)
v = velocity of flow, fps or cm per sec
p = density of the fluid, in lb per cu ft or g per cu cm (grams per cubic centimeter)
u = absolute viscosity of the fluid, lb per ft-sec, or centipoises

Once we have the Reynolds number, then we face a complication, or rather a matter for decision. It is this: Is our flow in the turbulent class, or is it in the streamline (sometimes called the viscous) category? The Reynolds number is supposed to indicate the proper category, but there happens to be a region in which the flow may flutter between the streamline and turbulent, and in these instances we may sometimes need to exercise our judgment.

The importance of the right category becomes clear when we observe that in conditions of viscous flow, the roughness of the pipe surface does not appreciably increase the friction, but with increasing turbulence, the surface roughness detains the fluid.

In either event, when we have a Reynolds number, we can use the friction chart (Fig. 1.2) to find a suitable friction factor for the Fanning formula.

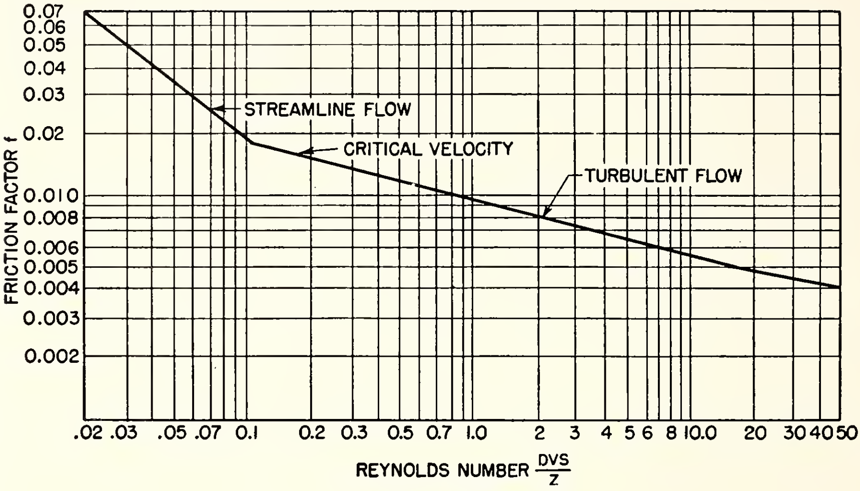

Fig. 1.2. The Reynolds number chart.

This number was first proposed by Prof. Osborne Reynolds in 1874. It was so little prized at the time that the name of Reynolds was hardly known, except perhaps by workers in the field of the flow of fluids.

Radiant Heating. If we hold a sheet of aluminum foil within an inch of our cheek, we immediately are conscious of the presence of heat. This heat actually is the heat radiating from our own flesh, reflected back to us by the aluminum foil.

Heat rays are invisible rays which move in straight lines. They are not affected by vacuum, but are checked, reflected, or absorbed by solid bodies, if these bodies are placed in their paths.

When we stand in the sun, or before a heated wall panel, or under a heated ceiling, the end effect is exactly similar to the feeling of heat we experienced from the aluminum foil; we are conscious of the impact of heat, not reflected this time, but originating in some distant source.

Now the significance of this phenomenon becomes clear when we realize that our feeling of heat occurs irrespective of the temperature of the surrounding air. The ensuing question then becomes: Why not warm our bodies with direct rays, instead of wasting heat on the whole contiguous space?

The employment of radiant heating, in the form of floor panels or pipes in the ceiling and walls, is the practical answer to this question. These radiant panels make possible full creature comfort with an ambient temperature of 66 F or even lower. This reduction in ambient temperature below the customary 72 F (or higher) is the theoretical measure of saving in the heating costs.

Actually, results from radiant heating can be less than we expect. It is not an idea without flaws. For instance, heat rays travel in straight lines; and if a chair, a table, or other object happens to come between us and the panel, the heat supply to our bodies is interrupted, and we may feel a chill. Again, rugs affect the heat emission from floors, and draperies affect that from walls; even the texture of the emitting surface must be allowed for.

The amount of heat transmitted by radiation is indicated by a solution of the Stefan-Boltzmann equation:

$$q = 0.174 \times e \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]$$

where q = quantity of heat transferred by radiation, Btu per unit of area per hour
T_1 = absolute temperature of the warmer body
T_2 = absolute temperature of the cooler body
e = surface emissivity, usually varying between 0.90 and 1.00

Some years ago, this phenomenon of heat transfer by radiation caused a condition that heating people did not understand; they called it "cold 70." Frequently we had 70 F inside with 0 F outdoors, and although everybody agreed that 70 F was warm enough for anybody, in these cases it literally left us cold. It was conventional then to ascribe the discomfort to the physical state of the sufferer; he was thought to lack red blood corpuscles, to be face to face with influenza, or perhaps just too far advanced in old age to be normal.

Actually the chill and shivers resulted from bodily heat loss to the low-temperature walls. From the surface of our skin, which varies from about 82 F on the arms to more than 90 F on the forehead, invisible heat rays are continuously pouring out in straight lines to every object that is simultaneously in the line of sight and lower in temperature than the skin. Up to a certain rate of heat transfer, say 250 Btu per hr total, we are not conscious of this loss of bodily heat; but cold walls can increase the rate materially, and then we are actually, unmistakably cold.

To warm us up under these conditions, we must provide an ambient air temperature in the room considerably above 70 F. If the walls are of light construction with a high coefficient of heat transfer, we may require an ambient temperature of 80 F before the occupants of the room will feel comfortable. This writer never designs for an ambient temperature lower than 74 F; and if long stretches of zero weather are expected, 76 F or 78 F are recommended. People can forgive almost anything from our heating systems except inadequacy.

Specific Heat. By definition, specific heat is the ratio of the heat required to raise the temperature of 1 pound of the given substance 1 degree Fahrenheit, and the amount of heat similarly required by 1 pound of water.

For example, the specific heat of water is 1, since 1 Btu will increase the temperature of 1 pound of water 1 degree Fahrenheit. For air, the specific heat is 0.24, which means that less than one-quarter of a Btu will change the temperature of a pound of air 1 degree, either up or down.

Specific heat for some substances changes if the substance changes its state or condition. For example, beef has a specific heat of 0.74 at normal temperatures, but if the beef is frozen, 0.30 becomes the correct value. As another illustration, when water changes into steam, its specific heat changes from 1.0 for the water to approximately 0.49 for the steam.

We must remember this characteristic of specific heat, and verify all values before we use them, in view of the conditions prevailing.

Table 1.6 provides some useful values of specific heat.

Steam. Since steam is an important factor in heating design and practice, we should understand thoroughly the characteristics of its various manifestations.

When water is transformed into steam, we call the end product "dry saturated steam." What we mean is that the steam temperature is exactly what it should be for the existing pressure, and that, moreover, it does not bear a suspended burden of water.

From time to time, steam is specified as to its "quality," which means that the amount of entrained moisture which will be accepted is strictly limited. Thus, if we call for "dry saturated steam of 98 per cent quality," we mean that moisture leaving the boiler with the steam must not exceed 2 per cent of the steam.

Water in steam is not desirable, particularly for engines and turbines, since it causes physical shock to the former, being noncompressible, and digs erosion and corrosion channels in the latter.

TABLE 1.6. SPECIFIC HEATS OF VARIOUS SUBSTANCES

Substance	*Specific Heat**
Air	0.24
Alcohol	0.46
Aluminum	0.23
Ammonia	0.52
Beer	0.90
Beef:	
Above 28 F	0.74
Below 28 F	0.30
Carbon:	
Below 100 F	0.16
At high temperatures	0.50
Carbon dioxide	0.25
Carbon monoxide	0.26
Copper	0.10
Gold	0.032
Gasoline	0.58
Hydrogen	3.470
Iron	0.16
Kerosene	0.50
Lead	0.32
Magnesium	0.25
Manganese	0.12
Mercury	0.033
Methane	0.62
Nickel	0.12
Nitrogen	0.25
Oxygen	0.22
Sulphur	0.21
Sulphur dioxide	0.155
Water	1.000
Water vapor	0.49

* *Note:* Specific heat varies with temperature. These are average values.

Superheated steam is dry saturated steam that has been reheated to some temperature higher than that which is normal to its pressure. For example, steam at 5 pounds per square inch gauge pressure (psig) has a normal temperature of 228 F, and if we pass this steam through a piping arrangement generally suspended in an intensely hot area of the boiler, it emerges unchanged in pressure but more or less elevated in temperature. The difference between this emerging temperature and the original 228 F is the amount of superheat we have added, expressed in degrees.

Superheat is of distinct advantage in delivering steam over long runs of piping, as the superheat is lost before the steam begins to condense, thus ensuring the delivery of steam rather than water at the point of destination. Superheat also improves the operating conditions for steam engines and turbines, by eliminating or at least reducing the impacts from slugs of water.

However, we do not want superheat in our radiators or convectors. The backbone of steam heating is the latent heat of vaporization, which is released of course when the steam condenses; but if superheat is present the steam resists condensing, and radiator capacities fall off. Researchers report that merely a few degrees of superheat will reduce heat emission up to 30 per cent. This writer has found the introduction of desuperheating sprays essential in some installations before satisfactory heating resulted.

Temperature Scales. Temperatures are expressed commonly in degrees Fahrenheit. The Fahrenheit scale is based on the boiling point of water at 212 degrees above what Fahrenheit assumed to be the absolute zero. On Fahrenheit's basis of reasoning, water froze at 32 degrees above the absolute zero.

Nowadays, we suppose the absolute zero to be approximately 460 degrees lower than Fahrenheit's conception of the term, but there is reason to suspect we shall be obliged to lower this conviction also, and before too long.

Less commonly but more sensibly, we use the centigrade scale, which ranges between the freezing point of water, called zero, and the boiling point of the same fluid, taken at 100 degrees.

To convert degrees centigrade into degrees Fahrenheit, we multiply the former by $\frac{9}{5}$ and add 32. Conversely, to change degrees Fahrenheit into degrees centigrade, we first subtract 32 and then multiply by $\frac{5}{9}$.

A third scale sometimes encountered, is the Réaumur gradation, which works on a range of zero for the freezing of water and of 80 degrees for the boiling temperature. This gives us a conversion multiplier of $\frac{9}{4}$ when changing Réaumur into Fahrenheit, plus 32. Changing Fahrenheit into Réaumur simply reverses the procedure, as in the case of the centigrade conversion.

Throw. The "throw" of air from a register or grille has the same meaning as "blow."

Ton of Refrigeration. A ton of refrigeration refers to 1 ton of ice, either being made or melted over a period of 24 hours. Since the fusion heat of ice is 144 Btu, a ton of refrigeration represents 2,000 pounds multiplied by 144, or 288,000 Btu in 24 hours.

On the 1-hour basis, a ton of refrigeration is equal to 288,000/24, or 12,000 Btu. The hourly basis is usually more convenient to handle in heating and cooling calculations.

Viscosity. There are two kinds of viscosity: kinematic and absolute. In either case, the term is used to identify an internal resistance of a fluid to movement, when this movement requires a readjustment in the relative position of its parts. An illustration of extreme viscosity is provided by the ancient expression "as slow as molasses in January." Here the resistance is so great that our cold molasses is practically a solid.

Absolute viscosity, by definition, is the *resistance* that must be overcome in order to move a plane surface of one square centimeter area at a speed of one centimeter per second, referred to another plane surface which is separated from the first by a layer of the liquid one centimeter thick.

The unit of this resistance, in terms of the cgs system (centimeter-gram-second), is the "poise." The force necessary to overcome the resistance of one poise is termed one "dyne," and a dyne is defined as the force needed to produce a velocity of one centimeter per second, when acting on a mass of one gram for one second of time.

The similarity of conditions employed in the definitions of both the poise and the dyne has tended at times to cause some confusion of understanding. In fact, absolute viscosity is sometimes even defined as a "force," with the measure of the force being the poise. For our own practices, our best plan is to recognize absolute viscosity at all times for what it is, a resistance pure and simple. From this point of view, the dyne as a unit does not enter our computations when we are designing a pipe line, determining friction factors, and so on.

Measurement of the absolute viscosity of fluids is a difficult and delicate procedure; consequently, for practical purposes we normally use the Saybolt viscosimeter and apply a factor to the Saybolt seconds, and thereby provide a satisfactory approximation. This step is necessary, because we must have the absolute viscosity of the fluid in order to be able to use the Reynolds number equation.

Fortunately, researchers have determined that a definite ratio exists between the absolute viscosity and density of a fluid; this ratio is called the "kinematic viscosity." The unit of kinematic viscosity is the "stoke." A standard stoke is created when the absolute viscosity value of 1 poise is divided by a density of 1 gram per cubic centimeter. Thus

$$\text{1 stoke of kinematic viscosity} = \frac{\text{1 poise}}{\text{1 gram per cubic centimeter}}$$

The ratio between kinematic viscosity and absolute viscosity is expressed by this equation:

$$\text{Kinematic viscosity} = \frac{\text{absolute viscosity}}{\text{density}}$$

This equation is fundamental, but it contains two unknowns: the kinematic viscosity and the absolute viscosity, each expressed in relationship to the other. To solve this situation, the researchers have produced other workable equations, such as this one by Nelson in "Petroleum Refinery Engineering"*:

$$\text{Kinematic viscosity} = 0.219t - \frac{149.7}{t} \qquad \text{centistokes}$$

t = viscosity, Saybolt seconds

With the kinematic viscosity known, we can substitute it in the equation for absolute viscosity and then readily determine the latter.

Kinematic viscosity we note is expressed in centistokes in this equation, a centistoke being $\frac{1}{100}$ of a stoke. When we use centistokes, then the absolute viscosity will come out in centipoises, a centipoise being $\frac{1}{100}$ of a poise. These are smaller units and easier to handle.

To summarize the viscosity discussion, what we need for our design problems is the absolute viscosity of a fluid in either poises or centipoises, and we must obtain this information either from actual measurement or through a conversion of the readings taken on a Saybolt viscosimeter. If we adopt the latter method, which is customary, then our first step is to change the Saybolt readings into kinematic viscosity, a simple procedure accomplished by means of the formula provided. The absolute viscosity results when we multiply the kinematic viscosity by the density of the fluid, and we thus obtain the denominator for the Reynolds number equation:

$$R = \frac{Dvp}{u}$$

u = the absolute viscosity of the fluid, centipoises

Illustrations of the working out of these formulas appear in Chap. 14, Fuels and Fuel Handling.

Viscosity Terms. While we have been discussing viscosity in terms of the centimeter-gram-seconds (cgs) system, it is also possible to solve viscosity problems using the English units of pound, feet, and seconds. Reynolds number equations frequently are set up on the English terms. Which system we use is immaterial; our usual care is for uniformity, *every* unit in the equation being chosen from the same system.

The English terms generally encountered are the "poundal," which is a unit of force and which identifies a force sufficient to produce a velocity of 1 foot per second when it acts on a mass of 1 pound for 1 second, and the "slug" and "geepound."

Both the slug and geepound refer to mass, and represent a mass weighing 32.1912 pounds at sea level.

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* McGraw-Hill Book Company, Inc., 1949.

In general practice, most of us will find the cgs units easiest to use. Some designers tend to mix their units, using feet and inches as the numerator of the Reynolds fraction and absolute viscosity in centipoises as the denominator. The result of course is a Reynolds number somewhat different than is obtained if the units are kept uniform throughout. This mixing practice certainly is not the most accurate thing to do, but the results appear to be satisfactory, indicating that practical tolerances are wide.

For those of us who practice uniformity, the following equivalents will be helpful:

1 centimeter = 0.3937 inch
1 centimeter = 0.0326 foot
1 square centimeter = 0.155 square inch
1 gram = 15.432 grains
1 kilogram = 2.2046 pounds, approximately

Questions

1. What is the difference between "absorbent" and "adsorbent"?
2. Define conductance and convection.
3. Discuss sound and the interpretation of decibel readings.
4. What is a degree-day?
5. Explain the dew point.
6. What is the relationship of density and volume?
7. What is the meaning of the term "enthalpy"?
8. What is equivalent direct radiation? Why is it important?
9. What danger must we safeguard against when we are discussing free air in relation to a fan?
10. Discuss the various heads encountered in pumping problems.
11. What is the relationship between humidity and the dew point?
12. What is latent heat? of evaporation? of fusion?
13. What is a mol? Where is it used to advantage?
14. Why is an understanding of the Reynolds number important to designers?
15. What is radiant heat?
16. Describe the two basic conditions of steam.
17. Name the three temperature scales commonly encountered.
18. What is a ton of refrigeration?
19. What is absolute viscosity?
20. What is kinematic viscosity?

CHAPTER 2

HOW HEATING LOADS ARE CALCULATED

Fundamental Decisions

Before we can set down any figures on which to base our calculations of heating loads, we must select certain design values as a point of departure. Five decisions are required, each of which we shall consider separately.

The Design Outside Temperature. In each community, it is customary to use an outside design temperature that has been more or less established over a period of years. However, despite its often lengthy history of use, adoption of this conventional temperature can be a dangerous procedure. The reason for this is primarily that the design temperature has been set by competition more often than not, and the established low is very apt to cause unsatisfactory heating in severe weather.

We must be particularly careful to avoid becoming overoptimistic as the result of several mild winters: climatic trends must be evaluated over half a century at least.

Table 2.1 provides information on the lowest recorded temperatures and the recommended design temperatures for various cities throughout the country. These figures are not immutable and must be used with local common sense. We all know that temperatures customarily vary throughout a single city. Some sections, as the result of elevation or prevailing winds, experience uniformly lower temperatures than do neighboring sections. This condition we must consider in designing for any specific installation.

Meanwhile, the prudent rule is this: When in doubt, always select the lower temperature. To say that we designed for the usual low temperature of the community is no excuse for an unsatisfactory job.

The Design Inside Temperature. For many years, it has been general practice to choose an inside design temperature of 70 F, measured at the so-called comfort line, 5 ft above the floor.

Unfortunately, this practice has provoked many complaints, and the usual business of pointing out a 70 F reading on the thermometer does not always appease the complainer. Many a contractor has been accused, bluntly, of supporting his heating guarantee by means of "fixed" instruments.

There are two reasons for this difficulty. First, temperatures at floor levels may be 5 or 6 deg below temperatures at the 5-ft line, and cold feet can cause a chill that travels throughout the body.

The second and far more important reason is found in the existence of "cold 70," a term discussed in Chap. 1, under the heading Radiant Heat. The curative treatment consists of checking convection losses from the body in order to compensate for the increased eduction by radiation. This can be accomplished in a simple manner: We design for an ambient temperature several degrees higher than the historical 70 F.

TABLE 2.1. LOW AND DESIGN TEMPERATURES FOR VARIOUS CITIES

City	Lowest temp. on record, F	Conventional low temp. for design, F
Portland, Me	−21	− 5
Boston, Mass	−18	0
Buffalo, N.Y	−20	− 5
New York, N.Y	−14	0
Philadelphia, Pa	−11	0
Washington, D.C	−15	0
Norfolk, Va	2	15
Charleston, S.C	7	20
Miami, Fla	27	35
New Orleans, La	7	20
Houston, Tex	5	20
El Paso, Tex	− 5	10
Albuquerque, N.M	0	0
Phoenix, Ariz	16	25
Los Angeles, Calif	28	35
San Francisco, Calif	27	35
Seattle, Wash	3	15
Butte, Mont	−34	−20
Minneapolis, Minn	−34	−20
Chicago, Ill	−23	−10
St. Louis, Mo	−22	0
Denver, Colo	−29	−10
Salt Lake City, Utah	−20	−10
Boise, Idaho	−28	−10

This writer never designs for less than 74 F in a residence, and if outdoor conditions are severe, 76 F to 80 F will not be found too high.

Unheated Spaces. The FHA minimum requirements for residential structures stipulate that unheated spaces shall be considered equivalent to outdoor exposures when computing heating loads.

This requirement appears a little stringent, since attics, cellars, and closed-off rooms, due to the absence of air movement and the impact of solar heat, rarely become as cold as the great outdoors.

The temperatures of unheated spaces can be computed of course, but the procedure is not often justified. Experience with existing construction, in the area where the heating installation is to be made, provides a sound basis for estimating temperature differentials. In cases of doubt, we can always fall back on the FHA requirements and be sure of a satisfactory result.

Some of the FHA requirements will be discussed later in this chapter.

Allowance for Exposure. There is no accurate substitute for judgment and experience in the evaluation of this factor. In mild climates, perhaps in city areas where the sweep of the wind is chastened, an allowance for northerly or western exposure is likely to be unnecessary. The usual range of allowance is between 5 and 15 per cent.

About Ground Temperatures. When calculating heat losses from cellar floors and walls, and through floors that are set directly on the earth, we are in immediate need of an earth temperature.

In most parts of the United States, winter ground temperatures fall below freezing only for the first 2½ ft under the surface. This means that temperature differentials through floors rarely are large, when compared with differentials across walls.

What temperatures we use must be selected on the basis of local conditions. Information generally may be obtained from the local agricultural agent; sometimes from the weather bureau or the public utility company; frequently from the water works office. Well-water temperatures offer a sound basis for calculations.

Coefficients of Heat Transmission

The fundamental heat-transmission coefficient for various types of building construction is

$$U = \frac{1}{\frac{1}{a} + \frac{x}{b} + \frac{x}{b} + \frac{x}{b} + \frac{1}{c}}$$

where U = over-all coefficient, Btu/hr/ft^2/°F difference in temperature
x = thickness, in., of each material that forms part of the construction of the wall, roof, floor
a = surface coefficient for the inside surface, generally taken as 1.65
b = conductance of each material, based on the thickness the design calls for (we will have as many x/b fractions as we have materials)

c = surface coefficient for outside surfaces, having an average value for conventional construction of 6.00, with a wind velocity of 15 mph

As an illustration of how this equation functions, we will consider a building wall composed of 4 in. of brick veneer, wood sheathing, building paper, and ½ in. of plaster on wood lath.

From Table 2.2, which provides conductances, we note that our 4 in. of brick has a conductance of 2.27; the sheathing and building paper, 0.86, and the wood lath and plaster, 2.50 Btu/hr/ft^2/°F temperature difference.

TABLE 2.2. CONDUCTANCES FOR VARIOUS MATERIALS*

Material	*Total Conductance for Thickness or Construction Indicated*
Air space, vertical ¾ in. or wider	1.10
Brick veneer	2.27
Stucco, for 1 in. thickness	12.50
Wood shingles	1.28
Yellow-pine lap siding	1.28
Gypsum board, ⅜ in. thick	3.70
Gypsum lath and ½ in. of plaster	2.4
Metal lath and ¾ in. of plaster	4.40
Plywood, ⅜ in. thick	2.12
Wood lath and plaster	2.50
Brick:	
Adobe, 4 in. thick	0.89
Common, 4 in. thick	1.25
Face, 4 in. thick	2.30
Clay tile, hollow, 4 in. thick	1.00
Concrete blocks, hollow, cinder:	
4 in. thick	1.00
8 in. thick	1.00
12 in. thick	0.80
Asbestos shingles	6.00
Asphalt shingles	6.50
Built-up roofing, ⅜ in. thick	3.53
Slate, ½ in. thick	20.00
Gypsum sheathing, ½ in. thick	2.82
Fir or pine sheathing, nominal ¾ in. thick	1.02
Fir sheathing, plus building paper	0.86
Still air, surface	1.65
15 mph wind, surface	6.00

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 6.

Inserting these figures in the equation, together with the surface coefficient fractions, we have

$$U = \cfrac{1}{\cfrac{1}{1.65} + \cfrac{1}{2.27} + \cfrac{1}{0.86} + \cfrac{1}{2.50} + \cfrac{1}{6.00}}$$

and

$$U = \frac{1}{0.606 + 0.44 + 1.16 + 0.40 + 0.167}$$

$$U = 0.361$$

Generally, we will seldom need to compute a coefficient, since the ASHVE tables provide us with solutions to the fundamental equation for all the conventional building materials and arrangements, and some that are not so conventional. We can abstract the coefficient from these tables in a fraction of the time required for the detailed calculation.

Tables 2.3, 2.4, and 2.5 have been abstracted from the ASHVE Guide.

Table 2.3. Coefficients of Heat Transmission through Various Walls*
(Expressed in Btu per Hour per Square Foot per Degree of Temperature Difference)

Type of wall	Thickness, in.	Plain walls	Plaster, ½ in. thick	Metal lath and plaster	Gypsum board ⅜ in. thick, furred
Solid brick	12	0.36	0.34	0.25	0.25
Hollow tile	8	0.40	0.37	0.27	0.27
Stone	12	0.57	0.53	0.35	0.34
Hollow concrete blocks, cinder	12	0.38	0.36	0.26	0.26
Brick veneer:					
On 8-in. hollow tile	..	0.34	0.32	0.25	0.24
On 8-in. concrete	..	0.54	0.50	0.33	0.33
On 8-in. cinder blocks	..	0.34	0.33	0.25	0.24
Wood sheathing under:					
Clapboard	..	....	....	0.26	0.26
Wood shingles	..	....	....	0.26	0.26
Stucco	..	....	....	0.32	0.31
Brick veneer	..	....	....	0.28	0.28

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 6.

Infiltration Allowance. Table 2.6 supplies information on the volume of cold air that leaks into a building around the windows and between the sashes. Door leakage may be estimated on the same basis of information, leavened by our opinions on the fit of the door and the size of the crack.

We must convert these inleaking volumes to Btu, in order to provide

TABLE 2.4. COEFFICIENTS OF HEAT TRANSMISSION THROUGH CEILINGS AND FLOORS*

Type of ceiling	Insulation		With flooring on top of joists	
	None	Blanket or bats, on or between joists, 3 in. thick	Single floor	Double floor
No ceiling	....	.....	0.45	0.34
Metal lath and plaster	0.69	0.093	0.30	0.25
Gypsum board	0.67	0.092	0.30	0.24
Wood lath and plaster	0.62	0.091	0.28	0.24
Plywood, 3/8 in. thick	0.59	0.091	0.28	0.23
No ceiling, with 3 in. concrete floor:				
Bare	....	.....	0.68	
Asphalt tile on concrete	....	.....	0.66	
Double wood floor on sleepers	....	.....	0.25	
Tile or terrazo on concrete	....	.....	0.65	
Concrete on ground	Use 0.10 whether floor has or does not have insulation, pending additional research			

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 6.

TABLE 2.5. COEFFICIENTS OF HEAT TRANSMISSION THROUGH GLASS AND DOORS*

Type of glass	Thickness, in.		
	Single	Double	Triple
Window panes	1.13	0.45	0.281

Smooth-surface glass blocks:	
Still air both sides	0.40
Still air one side, 15 mph on other	0.49
Ribbed-surface glass blocks:	
Still air both sides	0.38
15 mph on outside	0.46
Wooden doors:	
1 in. nominal thickness	0.69
1¼ in. nominal thickness	0.59
1½ in. nominal thickness	0.52

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 6.

TABLE 2.6. INFILTRATION VALUES FOR COMPUTING CRACK HEAT LOSSES*
(Expressed in Cubic Feet of Air per Hour per Foot of Crack)

Type of window	Wind velocity, mph		
	15	20	30
Double-hung wood sash:			
Average	39	59	104
Weather stripped	24	36	63
Rolled section, steel sash:			
Industrial	176	244	372
Architecturally projected	62	86	139
Residential casement	33	47	74
Hollow metal, vertically pivoted	145	186	242

* *Note:* The data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 8.

TABLE 2.7. WEIGHT AND VOLUME OF AIR AT VARIOUS TEMPERATURES

Temperature, F	Weight, lb/cu ft	Volume, cu ft/lb
−30	0.092	10.82
−20	0.090	11.07
−10	0.089	11.326
0	0.087	11.58
10	0.084	11.83
15	0.083	11.96
20	0.0825	12.08
30	0.081	12.34
50	0.078	12.84
75	0.074	13.47
100	0.071	14.11
150	0.065	15.37
200	0.060	16.63
300	0.052	19.14
400	0.046	21.66
450	0.043	22.90
500	0.041	24.18
600	0.037	26.70
700	0.034	29.22
800	0.032	31.74

for this chill in our load calculations. The conversion is accomplished by solving the following equation:

$$\text{Btu} = \frac{\text{cu ft per hr inleakage} \times Td \times 0.24}{\text{cu ft per lb of inleaking air at outside temperature}}$$

Where Td = degrees temperature difference between inside and outside air

0.24 = specific heat of air

Table 2.7 provides us with information on the weight and volume of air at various temperatures.

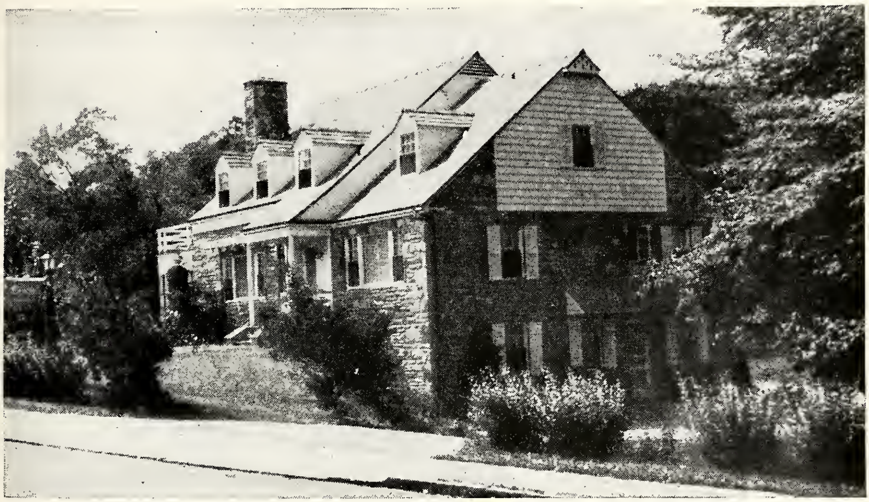


Fig. 2.1. A large residence to be heated by mechanical warm air.

In measuring the crack, we follow the perimeter of the portal, whether door or window, and add cross joints, if any. We do not charge crack losses to all the doors and windows, however, since inleakage on one side of the room must be balanced by outleakage on the other. The usual practice is to allow for crack on one wall only, although here, too, individual judgment is needed. From time to time we encounter special conditions, such as the frequent opening of a door, that indicate infiltration will occur over a wider than usual area; consequently we must adjust our estimates accordingly.

Typical Calculating Procedures

The first step is to tabulate the design values with which we shall work. As an example of how these data are set down and used, a reproduction of procedures used in a typical job (Fig. 2.1) follows:

Location: Haverford, Pa. (near Philadelphia).

Building: Combination residence and doctor's office, incorporating two full floors and unheated attic, built into a hillside so that the basement is partly excavated, partly aboveground. Basement houses the doctor's waiting room, examination room, and office in a separate suite, a three-car garage, the heater room, and a recreation room.

Construction: Stone walls 18 in. thick at basement elevation, 16 in. thick above the basement. Wood siding in certain areas, concrete flooring at ground level, wooden floors on other levels, bat insulation over the second-floor ceiling.

Basic Design Factors:

Outside design temperature	0 F
Inside design temperature	75 F
Unheated space, attic	35 F
Ground-hall passageway	60 F
Garage	45 F
Ground temperature	30 F
Allowance for exposure	None. House is protected by trees on north and west sides

Heat-transmission Factors: From the tables of heat transmission through various types of building construction, the following coefficients have been selected:

For the 18-in. stone wall	0.34
For the 16-in. stone wall	0.39
For the wood siding	0.26
For the concrete floor	0.10
For the insulated ceiling	0.20
For the wooden floors	0.34
For wooden doors	0.69
For glass	1.13

Infiltration 20.2 Btu per lin ft of crack

The physical data for each room are then tabulated in this manner:

1. Waiting room	10′0″ × 17′4″ × 8′0″ ceiling
Outside wall	(10′0″ + 17′4″) × 8′6″ (*Note:* 6 in. has been added to ceiling height to allow for ceiling thickness)
Gross area, outside wall	232 sq ft
Glass, 3 windows and 1 door, total area	40 sq ft
Net outside wall	192 sq ft
Inside wall (against garage)	111 sq ft
Floor, 10′0″ × 17′4″, total area	173 sq ft
Crack, 2 windows, 36 lin ft 1 door 23 lin ft	59 lin ft

After every room and space has been measured, we are ready to apply our coefficients and determine the heat loss. This is done on a summary form in a manner similar to the following:

Room	Surface	Area, sq ft	Temp. diff.	Coef.	Total, Btu/hr
1. Waiting.........	O.W.	192	75	0.34	4,896
	I.W.	111	30	0.34	1,132
	Glass	40	75	1.13	3,400
	Floor	173	45	0.10	779
	Crack	59 ft	75	20.2 Btu	1,192
Total for waiting room......................................					11,399

The coefficients are applied to each room, as illustrated by this case of the waiting room. The total of all room loadings then becomes the calculated heat loss of the building.

This total loss is net, and in sizing the heater or boiler we will add additional Btu to meet the pickup demand and also in some instances to supply domestic hot water. These procedures are discussed in later chapters which deal with sizing the equipment and designing the chimney.

Minimum FHA Requirements

In general, the Federal Housing Administration (FHA) minimum requirements rest on a sound and conservative engineering viewpoint. Occasionally, current practice will term them overconservative, a criticism that in many individual instances is justified, but certainly the country-wide commitment of public funds in enormous amounts is deserving of extensive protection.

The minimum requirements vary with the type of building involved, and whenever FHA financing occurs on a construction project, we should obtain the directives applying. Here, for example, are a few notes taken from the "FHA Minimum Property Requirements for Properties of Three or More Living Units:"

1. *Fuel storage.* Provide storage for not less than one month's requirements.
2. *Outside design temperature.* Secure the requirements from the FHA for the locality.
3. *Inside design temperature.* 70 F minimum.
4. *Infiltration.* Allow rate of one air change per hour.
5. *Total heat loss.* Calculated loss must not exceed 60 times the floor area in square feet (this is a check on construction).
6. *Coal-fired equipment.* Do not install above the first floor.
7. *Boiler or furnace.* This equipment must be mounted on one of the following:

   a. A fireproof floor
   b. Noncombustible flooring
   c. Fireproof slabs
   d. Four inches of hollow tile
   e. Asbestos panel, ¼ in. thick, covered with 24-gauge sheet metal

8. *Overflow heaters.* These are floor furnaces and ductless space heaters. Their heating range extends through one doorway only. The load on any one of these is not to exceed 45,000 Btu per hr.
9. *Automatic control.* Automatic control is a must on all systems that serve more than eight living units.
10. *Pumps.* Main pumps and circulating and condensate pumps must be provided in duplicate if the system serves more than 25 living units.
11. *Boilers required.* Two boilers at least must be furnished for all installations which include 60 or more living units.

Solar Factor

Unless we are building a strictly solar house, with the design objective of harvesting all the sun heat possible, we do not make any allowance for a sun bonus when calculating the building load. We have no guarantee that the sun will shine when and where we need the heat.

Experiments are now being made in various parts of the country to find means of collecting the sun's heat in some form of storage media for later use during the night hours, or on overcast days.

To date, these sun heat designs have not reached the commercial market, and their employment must be regarded as a practice not yet established.

Questions

1. Name the five fundamental decisions that must be made before the actual calculation of a heating load can be undertaken.

2. What is the coefficient of heat transmission?

3. How do you estimate the air infiltration from crack?

4. What is the winter ground temperature in your locality?

5. What is the customary outdoor design temperature in your locality? Is it satisfactory? How many degrees is it higher than the lowest recorded temperature? What is the *average* minimum low temperature for your locality?

6. Estimate the heating load for your own house, setting up the data in the form shown in this chapter.

7. Do you consider the FHA minimum requirements to exemplify good practice?

8. Why should a boiler be set on hollow tile?

CHAPTER 3

DESIGNING A STEAM HEATING SYSTEM

Basic Considerations

After the hourly Btu load required by any structure we are planning to heat has been calculated, the next step is to select the kind of system best suited to the building design and location.

We start, apparently, with plenty of choice among the numerous forms of steam, hot water, warm air, and electric heating, but in practice no one of them will meet all our requirements all the time. Knowing this, we become obliged before making a decision to familiarize ourselves with the advantages and limitations of each; until we do this we are not qualified to put a finger on any one system and say "this is best." Therefore in this chapter we will begin our examinations with steam heating design, and observe for ourselves why this method of producing winter comfort sometimes seems to be more temperamental than the weather.

Nature of Steam. Steam is a continual paradox. For instance, if we see steam, it is not steam, but water vapor. Again, for steam to serve us best, we must destroy it by condensing it into water, but if we have too much water, the steam refuses to travel satisfactorily through the pipes, and gives us little heat and a great deal of noise, called "water hammer."

A third paradox is that if our steam is too hot as the result of superheating, we get less heat, because the steam resists condensing and hoards its burden of heat within itself.

This apparently "zany" behavior is created by natural laws which we must recognize and understand if our steam heating system is to be efficient and docile in its operation. For example, under natural laws, water boils at 212 degrees Fahrenheit at an atmospheric pressure of 14.7 pounds per square inch (psia), but the same laws demand that it boil also at 180 degrees Fahrenheit if the pressure is reduced to 7.51 pounds per square inch absolute (psia). In fact, water actually will boil at 32 F under a pressure of 0.088 psia; thus we can have steam at almost any temperature we desire.

We use our knowledge of this law to maintain any temperature in our radiators that we wish, simply by hooking up a vacuum pump on the return piping. Figures 3.1 and 3.1*a* illustrate a type of vacuum pump well known in heating activities. The pump reduces the pressure in the piping to whatever subatmospheric pressure corresponds with the steam temperature we wish to maintain, and the water in the boiler obediently releases the steam at the reduced temperature.

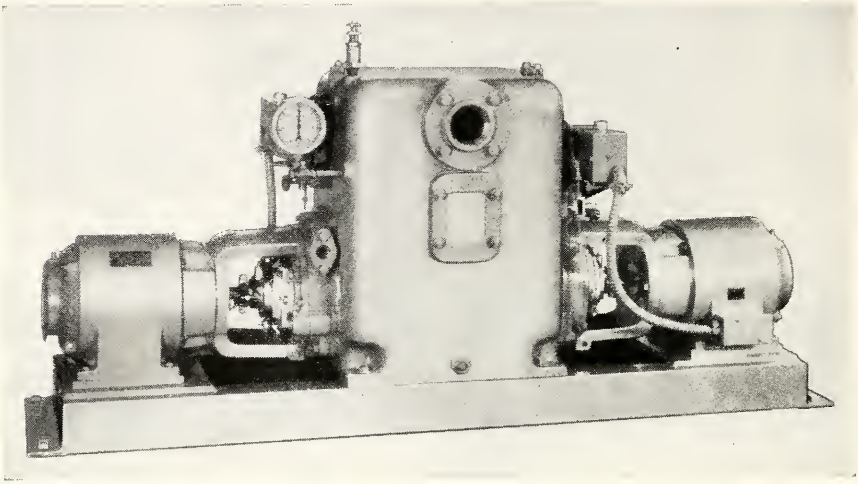


Fig. 3.1. Vacuum return pump with two motors to provide service dependability. (*Courtesy of Nash Engineering Co.*)

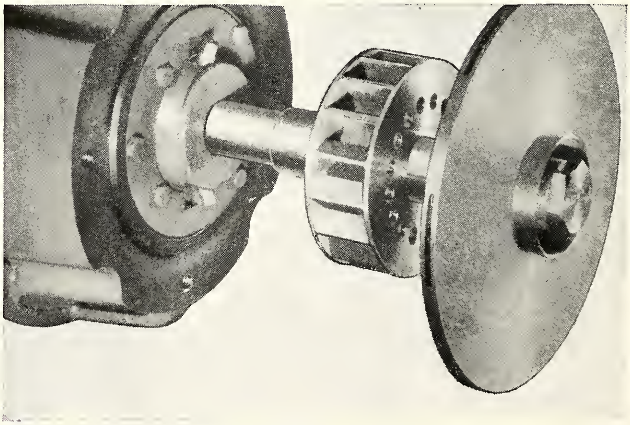


Fig. 3.1*a*. Pump rotor and air handling wheel on common shaft for vacuum return pump. (*Courtesy of Nash Engineering Co.*)

The value of this flexibility is highest in the spring and autumn, for with these low temperatures we take off the chill of a room very nicely, and save fuel doing it.

Importance of Drainage. We all know that the laws of steam demand that it must yield up its latent heat in order to condense. This latent heat is our profit, being distributed from the surface of our radiators, but the water of condensation so formed becomes a most important designer's problem. We dare not leave the water in the radiators, for then the steam refuses to enter; and we cannot permit it to lag in the piping, or collect at corners, because the movements of steam are blocked or at least interrupted thereby. The behavior of a dog and a strange cat locked in an attic approximates the behavior of steam and water mixed carelessly in a heating system; neither willingly tolerates the other.

Superheated Steam. Another law that we use to our advantage is the one that governs the behavior of superheated steam in matters of heat transfer. Superheated steam acts like a dry gas, and the transfer of heat from a dry gas to a dry metal surface occurs notably slower than does the transfer of heat from the same gas to a liquid. In other words, we need a good heat conductor between the gas and the metal, such as a film of water, and it is difficult to make superheated steam provide this film.

By contrast, the steam that normally arises from the water in our boiler bears a burden of moisture that sooner or later wets the surfaces of the pipes and radiators through which it passes, and this wetting becomes the film that accelerates the transfer of heat, causes the steam to condense, and dumps the latent heat so released into our laps, or wherever else we want it.

To break this contrast into usable figures, let us assume that we have a radiator rated to condense 25 lb of steam per hour when supplied with saturated steam at 212 F. Now we will make the steam go through a superheater in which its temperature is raised to, say, 225 F. These 13 deg of temperature difference represent approximately 6½ Btu per lb; we can verify this figure by glancing at Table 1.1, which details the properties of steam.

However, 25 lb times 6½ Btu per lb means that about 162 Btu of superheat must be disposed of before condensation is completed in full, and this causes a delay which is stretched out by the reluctance of heat to abandon the superheated steam for the dry wall of the radiator. In brief, we are not likely to condense 25 lb of this superheated steam in the elapsed hour. The heat output of the radiator is therefore reduced.

As far back as 1901, there is recorded in the *Transactions of the American Society of Heating and Ventilating Engineers* (ASHVE) tests made by Prof. R. C. Carpenter on the comparative heat emissions of saturated and superheated steam in cast-iron radiators. According to these tests, an 18-in. radiator, when served with steam bearing 32 deg of superheat, showed a reduction in heat emission of approximately 22 per cent.

Later researchers have verified the reduction in principle, if not in the same degree, and manufacturers of heat exchange coils for heating work estimate that the over-all reduction approximates 7 to 8 per cent.

The exact reduction is not essential for us to know in practical heating work, for there are so many other variables that the effect of superheat on economical heating becomes submerged in the multitude. We *are* concerned, however, with the importance of superheat in checking heat losses in our steam lines between the boiler and the radiators, particularly if these steam lines follow a long road with opportunities for heat loss along the way. Good insulation, as every district heating steam company knows, is not the whole guarantee of economical steam transmission. We desire to keep the heat in the steam, not merely inside the insulation, and superheat with its dry gas nature is our valuable ally to this end. Every Btu that fails to go through a customer's meter represents fuel and labor that are lost en route (largely to the water removal traps), and if enough of these Btu are lost, the customer gets too little steam, and the steam factory goes out of business.

We must consider the advantages and disadvantages of superheat very carefully when we take on the design of a large system. Overheated steam in the radiators actually can be far cheaper than underheated steam in a long pipe line. We even save money on trap installations along the route.

The Air Factor. Air in a heating system which is not designed to use air as a heating agent is strictly an unwanted presence. It accumulates in the high points to which it is pushed by the steam, and there it invariably makes a successful last stand, using its bulk to block steam circulation and immobilize parts of the radiators.

In addition, some permeates the condensate from which it later escapes, when boiling occurs, as a corrosive oxygen which destroys our pipes and boilers.

Eliminating air from a steam system means that we must supply vent valves at strategic points, generally high points. We shall see more of this venting business when we discuss the design of specific systems later in this chapter.

Some Rules for Steam Design. We now know enough about the paradoxes of steam to set down a few rules for our guidance in the problems of design. Here they are:

1. Lay out all pipe lines to avoid conflict between steam and the condensate.
2. Design against condensation in the steam mains. We may waive this rule if we plan to use the pipes as heat-emitting surfaces.

3. Allow only condensate and not steam to flow back from the radiators.
4. Keep air out of the pipe lines and radiators.

Each of these rules is subject to elaboration in its application to specific systems. Thus a one-pipe system demands an application technique quite different from that of a two-pipe system; a dry return presents its own especial needs as compared to a wet return, although, basically, they may be the same systems.

One-pipe Design

Figure 3.2 illustrates as simply as possible the relative arrangement of boilers and radiators in the one-pipe steam system. Steam rises from the boiler to the radiator with the condensate falling back to the boiler by gravity through the same pipe.

This arrangement apparently violates the first rule of steam design which tells us to avoid conflict between the steam and the condensate, but actually if the pipe is large enough and run properly there will be no conflict. The steam and water will accomplish their respective missions in tranquility.

This matter of pipe size is an essential consideration in designing a one-pipe system; if we undersize the piping, we are ensuring ourselves trouble.

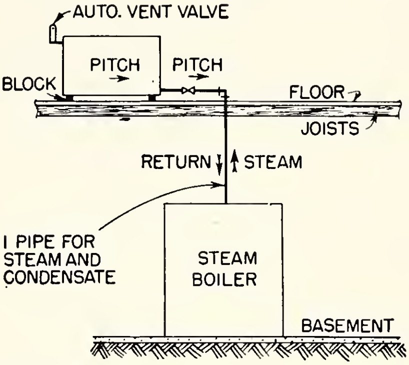


Fig. 3.2. Diagram of one-pipe steam system.

One-pipe Advantages. The one-pipe system has the following advantages: (1) it is the cheapest to install of all the steam systems; (2) it is simple in principle; (3) properly designed and installed, it works beautifully.

One-pipe Disadvantages. The disadvantages line up this way: First, incorrectly designed or installed, it becomes a top-ranking headache. Second, because of probable traffic troubles between the steam and water, it is not suited for high buildings. In fact, three-story structures appear to be a prudent maximum for the designer and two stories is even better. Third, because of gravity considerations in the condensate returns, it cannot be used readily with low radiators or underground returns.

Laying Out a Satisfactory One-pipe System. Suppose that we study the design shown in Figs. 3.3 and 3.4 which present the plan and elevation of a two-story house with basement.

Beginning with the plan, we note that the single pipe supply main is divided into two headers, one for each side of the building. Some designers would be satisfied with a single main, but such an arrangement imposes long travel on the steam in order to reach the last radiators, and during this drawn-out journey our steam meets ever-increasing opposition and crowding by the returning condensate. Trouble then can be avoided only by an intermediate dripping of the main or by a substantial increase in pipe size, and either of these expedients quickly increases the cost to a point that makes the two mains cheaper.

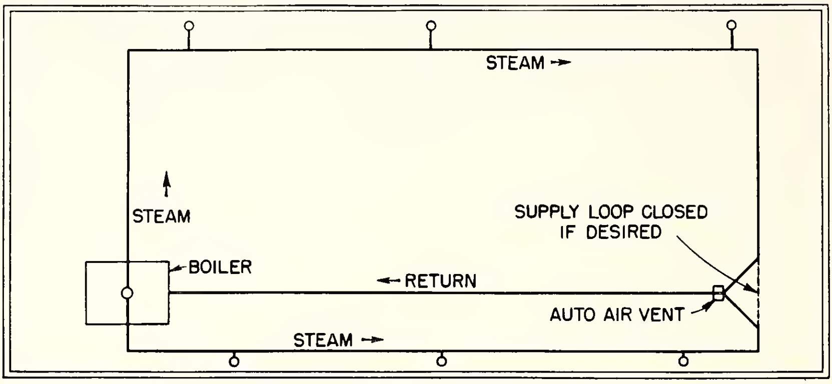


Fig. 3.3. Plan of two-story and basement house.

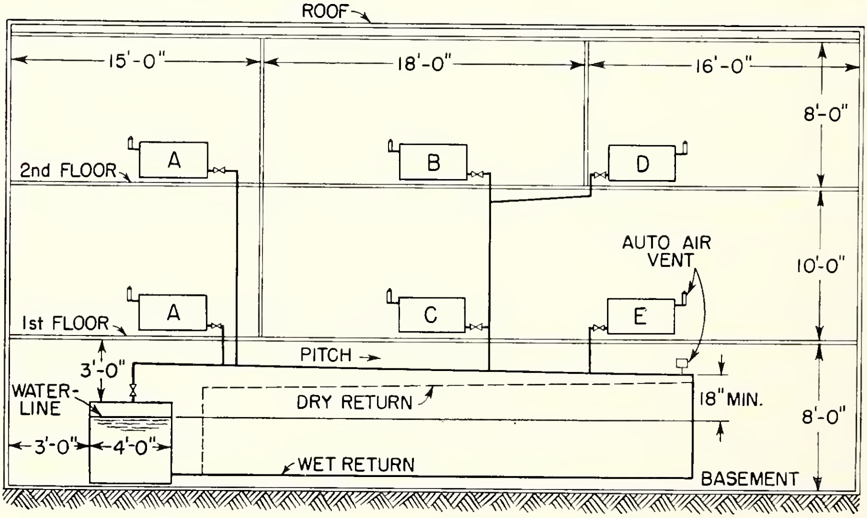


Fig. 3.4. Elevation of two-story and basement house.

Now examining the elevation, we observe that the basement supply header pitches away from the boiler, so that nowhere in this header is the steam in directional conflict with the condensate. That means we can expect quiet and peaceful circulation in this portion of the system, at least. Next we note that a minimum difference is specified between the water line in the boiler and the height of the returning condensate in the return piping. To understand why this difference is necessary, we might look at Fig. 3.5. Here is a U tube, made of glass so that we can see the water inside it and watch the water's actions. First, the water is still, and quite level in both legs of the U tube. Now if we blow a little in one leg of the tube, the water in the other leg rises, being pushed up by the pressure of the water we are blowing against. The harder we blow, the higher goes the water, and if our wind holds out, we can blow the water right out of the tube.

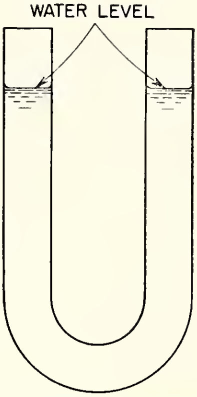


Fig. 3.5. Principle of the U tube.

Approximately the same phenomena occur in a steam heating system, with the steam pressure generated in the boiler taking the place of our lusty lungs, and the return piping acting as the low-pressure leg of the U tube. However, as our heating system is really a closed loop, the pressure should equalize itself throughout, but does not, because of friction losses between the boiler and the return piping. The farther we depart from the boiler, the less becomes the steam pressure, and by the time we reach the end of the supply piping and the point where the return begins, we have little or none of the starting pressure left.

Naturally this situation affects the level of the water in the return. At the boiler, the steam pressure exerts itself downward as well as upward, trying to force the water out of the boiler and into the return piping. The depreciated steam pressure above the return at the point where the return takes off from the supply system is unable to stop this tendency of the water to rise, and thus the water actually climbs up in the piping to a level somewhat above the water line of the boiler. How high above the boiler water line it goes depends on the difference of pressure. For instance, a difference of approximately 7 oz will force the water 12 in. higher in the return.

We are not likely to design a system with as much as 7 oz pressure loss in the steam piping, but strange things happen to piping as the years go past. Sediment and foreign bodies collect here and there, rust develops and offers growing resistance to the movement of the steam, and last but not least sags in the piping produce water and air pockets where we least want them. All these mean a growing friction loss, and after a few years we may have not only 7 oz of loss, but double that, in some particularly bad cases. Just to be reasonably safe, we will keep our steam header at its lowest point at least 18 in. above the water level of the boiler. This will allow the friction loss to climb all the way to 10 oz before water begins to back up in the steam header.

And what happens if water backs up into our steam header? Sometimes nothing. More often, the steam circulation is checked or stopped. One or more radiators will refuse to heat and will maintain that stubborn attitude until the water block is removed. We may have some hammering, suggestive of the anvil chorus. Sooner or later the owner will tell his friends that as designers, we do not know our business. Any one of these difficulties is one too many.

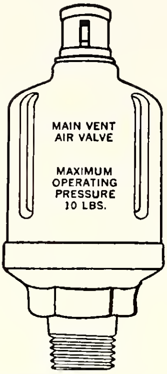


Fig. 3.6. Automatic vent valve for return piping. (Courtesy of Hoffman Specialty Co.)

One thing we should remember in connection with this water-level problem. The water line of a boiler tends to rise when the boiler is steaming. We can see this phenomenon anytime by putting a pot of water on the stove and watching the rise of the water as it begins to boil. This swelling is caused by the formation of thousands of steam bubbles within the body of the water that lift the water with them as they come to the surface. The steaming level therefore is the level from which we should count the inches to the bottom of the steam header at its lowest point.

Air Valve. This is an automatic contraption that we will mount on the high point of the return, and as far above the piping as the ceiling will permit. This valve will exude the air which is being pushed along by the steam and water; but when the steam or water arrives at the point of exit, the heat operates a thermostatic element, and the valve closes. These valves are fitted often with internal floats to close the vent, and some of them contain a bellows which operates on vacuum and prevents reentry of the air when the steam and water subside.

Figure 3.6 illustrates a typical steam system air vent valve.

One-pipe Runouts and Risers. Studying the elevation of Fig. 3.4, we observe that upper floor radiators sometimes are served by a single riser, while in other places several radiators appear on the same riser. Either arrangement is correct and either will work satisfactorily, provided the riser pipes are large enough and adequately pitched. Casual views of the layout suggest that several radiators on a single riser multiply the chances for trouble; actually the individual connection is equally dangerous if carelessly sized.

Figure 3.7 shows the way we should connect the risers to the header to safeguard the operation against trouble. First, we have a 45 deg take-off rather than a top take-off, because with the latter, condensate falls directly across the steam moving in the header, thus laying a water veil across the circulation. With the 45-deg take-off, water from the riser simply trickles down the side walls of the header whence it finds its way to the return.

Next we note that the runout from the header to the riser has a substantial pitch. Sometimes after a system has been functioning well for years, the building settles and either neutralizes or reverses the pitch. When this happens, the radiators on that branch are sealed off by the accumulation of water and refuse to heat. When investigating trouble, a check of the runout pitch often brings us surprisingly happy results.

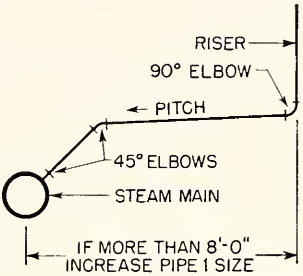


Fig. 3.7. Riser take-off from steam header.

Return Piping for the One-pipe System. The solid line along the floor of Fig. 3.4 shows the position we like for our return piping in the basement. This arrangement gives a good water leg and keeps the piping wet at all times. Piping that is always submerged resists corrosion better than does piping which is alternately wetted and dried.

This design of the return is known as a "wet return."

The dotted line in the illustration shows an alternative way of getting the condensate back to the boiler. This arrangement keeps the return high above the floor until it reaches the boiler room. Since it is not filled with water, but rather carries a trickle of water straight to the boiler without holding any back within itself, this design is called a dry return.

Dry returns are always above the water line of the boiler if they are to stay dry. Generally a dry return is used when practical difficulties or interferences prevent our laying the return pipe on the floor. However, some designers favor a dry return as being more accessible.

How to Size the Piping for a One-pipe System

General Considerations. To size piping correctly, we must consider any given system as being made up of four components, each of which must be evaluated for its own peculiar characteristics. Here are the four components:

1. The steam header
2. The runouts

3. The risers
4. The returns

Table 3.1, which has been developed by the American Society of Heating and Ventilating Engineers, tells us how much load we may impose safely on our piping. The unit of length for the piping is 100 lin ft, including allowances for fittings mounted along the way. This table is for use with headers and risers exactly as written, but for runout piping we must consider the influence of length on circulation through the runout. If the runout is more than 7 ft long, we must select a pipe one size larger than shown in the table, and we must never, if we value our sleep at night, use a runout smaller than 1 in.

Table 3.1. Capacities of Steam Mains and Risers, in Square Feet of Equivalent Direct Radiation (EDR)*

Pipe size, in.	Condensate flowing with steam pressure drop per 100 ft, oz			Condensate against the steam		
	½	4	8	2-pipe vertical	Horizontal	1-pipe supply risers
¾	.....	.....	.....	30	.....	25
1	39	111	157	56	34	45
1¼	87	245	346	122	75	98
1½	134	380	538	190	108	152
2	273	771	1,091	386	195	288
2½	449	1,270	1,800	635	395	464
3	822	2,330	3,290	1,130	700	800
4	1,740	4,910	6,950	2,040	1,700	1,520

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide, Chap. 23. For radiator runouts longer than 7 ft use the next larger pipe.

To size our return piping, both wet, dry, and in risers, we refer to Table 3.2, which also has been developed for the benefit of engineers by the ASHVE. As in the case of the steam piping, we must calculate the equivalent length of the run by adding together the actual length of the pipe and the allowance for fittings as tabulated in Table 3.3.

Sizing a Specific Job. Now suppose that we take the house shown in Figs. 3.3 and 3.4 and size the piping just to see how it is done. We shall assume that the radiator loads are those given in Table 3.4.

The boiler for this load is just a little fellow, beautifully jacketed in the modern style. We shall assume that it has a water line 4 ft 0 in. above the floor. If we add 18 in. to this water line for a pressure differential between the boiler and the return, we still have 2½ ft of space to

TABLE 3.2. CAPACITIES OF RETURN MAINS, IN SQUARE FEET OF EQUIVALENT DIRECT RADIATION*

Pipe size, in.	Pressure drop, oz, mains							
	½		4			8		
	Wet	Dry	Wet	Dry	Vacuum	Wet	Dry	Vacuum
¾	.....	.....	......	.....	800	.....	.....	1,130
1	500	248	1,400	460	1,400	.....	.....	1,980
1¼	850	520	2,400	962	2,400	.....	.....	3,390
1½	1,350	822	3,800	1,510	3,800	.....	.....	5,370
2	2,800	1,880	8,000	3,300	8,000	.....	.....	11,300
2½	4,700	3,040	13,400	5,450	13,400	.....	.....	18,900

Pipe size, in.	Pressure drop, oz, risers			
	½	4		8
	Dry	Dry	Vacuum	Vacuum
¾	190	190	1,400	1,980
1	450	450	2,400	3,390
1¼	990	990	3,800	5,370
1½	1,500	1,500	8,000	11,300
2	3,000	3,000	13,400	18,900

* *Note:* Data in these tables have been abstracted with permission from the 1948 ASHVE Guide, Chap. 23.

TABLE 3.3. RESISTANCE OF FITTINGS IN EQUIVALENT LENGTHS OF STRAIGHT PIPE FOR STEAM, IN FEET*

Pipe size, in.	90-deg ell	Side outlet tee	Gate valve	Globe valve	Angle valve
½	1½	3	½	14	7
¾	2	4	½	18	10
1	2	5	½	23	12
1¼	3	6	½	29	15
1½	3½	7	1	34	18
2	4½	8	1	46	22
2½	5	10	1	54	27
3	6½	13	1½	66	34
4	9	18	2	92	45

* *Note:* Compare these resistances with those given in Table 14.16. The two tables have been computed by different methods and from different experimental reports. The engineer must exercise his own judgment in each case.

TABLE 3.4. RADIATOR SCHEDULE

Symbol letter	Number of radiators	Heating surface, sq ft EDR	Total load, sq ft
A	2	100	200
B	1	60	60
C	1	45	45
D	1	120	120
E	1	30	30
Total load. . .	. . . .	. . . .	455

the ceiling, which is plenty for pitching the header. So we will change these 18 in. to ounces pressure drop with ease by use of the following equation:

$$1\frac{1}{2}' \times 16 \text{ oz} \times 0.434 = 10.4 \text{ oz}$$

This equation, a very useful one to remember, is based on the established premise that a column of water 1 ft high exerts a pressure at the base of the column of 0.434 lb.

Now we have 10.4 oz to lose in pipe friction, getting our steam around the bends, through the elbows and valves and, also, lest we forget, getting the water back to the boiler. Of these 10.4 oz, we will arbitrarily assign 6.4 to the steam side of the system, which embraces the bulk of the piping, and allow 4.0 oz to the returns. We may need to change this arbitrary division of losses after we count up the equivalent pipe lengths, but for the moment it lets us start nicely on the use of the sizing guide, Table 3.1.

As we find no column based on a 6-oz pressure drop, we will move to the 4-oz column, the only safe thing to do. Here we find that a 2-in. main with the steam and condensate flowing in the same direction is adequate for 771 sq ft of equivalent direct radiation. However, we plan to serve only 455 sq ft with this main, and therefore briefly consider using a 1½-in. pipe, but at 4 oz drop the 1½-in. pipe will handle only 380 sq ft. A 2-in. pipe then will be the size of our header.

We next consider the runout from the header to the riser. At this point we have a load of 225 sq ft spread over three radiators, and now, looking at the column for radiator and riser runouts, we make a surprising discovery: our runout will be of larger pipe than the main. This relationship is not met every day, and it is proffered in this case to illustrate first, that it can happen, and second, to emphasize the importance of sizing runouts generously. For the riser itself, we note that a 2-in. pipe will handle 288 sq ft, and is therefore the size we need, from header to the *D* radiator branch.

From the practical standpoint, observing that we must have a 2½-in.

runout, to size the main itself at 2½ in. would be desirable rather than to keep it at 2 in. The general appearance is more workmanlike, and the downflow of condensate from the runout finds its own level in the bottom of the main with less conflict.

The radiator runout from the riser to radiator *D* with a load of 120 sq ft will require a 2-in. pipe, according to Table 3.1. As the length of this runout is approximately 10 ft, the size should be increased to 2½ in. (see footnote of Table 3.1); but as such an arrangement is clumsy, top-heavy, and generally of a poorly engineered appearance, we will examine several other factors before making a decision.

Suppose that we look at the radiator first. A conventional 5-tube radiator, 25 in. high and 50 sections long, will provide 120 sq ft of heating surface. These 50 sections stretch out over a distance of 87½ in., which is no less than 7 ft 3½ in. of our available wall space.

This, we quickly gasp, is too much. No woman wants a room with a 7-ft radiator dominating the furniture arrangement. Nor are we, as engineers, satisfied with this arrangement, for so much concentrated radiation is quite likely to divide the room into hot spots and cold spots. We must distribute our heating.

Therefore we break up the big radiator into two smaller ones, and set them under the windows where they will do the most good, and to each we give its own runout. What size are these new runouts? The table says 1¼ in. each, but as the distance is over 8 ft, we will use 1½ in.

Thus we finally come to sizing the longest run of our one-pipe steam system. We shall have a 2½-in. header, a 2½-in. riser runout, a 2-in. riser, and a 1½-in. radiator runout.

Pressure Drop. With our sizes tentatively selected, we are now ready to compute the pressure drop through this run. The word "tentatively" is used because, if our pressure drop turns out to be greater than the 6.4 oz allowed, we shall need to do at least one of several things: (1) increase the size of the piping; (2) design the returns for a lower loss than originally planned; (3) increase the height of the steam main above the boiler water line. In actual jobs, we may find any one of these choices ruled out by existing conditions, for example, a low ceiling may compel us to reduce our 18-in. pressure differential rather than to increase it, or an involved and long-wandering return can build up a disturbing resistance. In fact, the first layout and sizing of a steam heating system is always tentative; if we hit the ultimate design right off, our good luck hints of either miracles or black magic.

In figuring the pressure drop, we must take the equivalent length of our piping, rather than merely the actual distance, because every time steam or water goes through an elbow, a tee, or a valve, it meets with a resistance far greater than is suggested by the shortness of the journey through the fitting. Table 3.3 provides us with data for converting these fitting resistances into equivalent lengths of straight pipe.

We might observe here that all authorities will not agree with the values shown in this table. Manufacturers and research laboratories have been testing flow through pipe fittings for years, and the results are so varied that some designers simply add 50 per cent to the actual pipe length for fittings, and let it go at that. Normally, this 50 per cent is more than ample for safety, as we shall see in back-checking this particular problem (Table 3.5). In the tabulation, we are assuming a loss through the radiator valve of 34 ft (taken from Table 3.3). This is a safe, perhaps overcautious decision, but one that we will not regret. Also, we observe, that the total fittings allowance is 55 ft, whereas, if we had added 50 per cent of the total length for this factor, the increment would have been 65 ft.

TABLE 3.5. EQUIVALENT LENGTHS OF PIPING

Pipe section	Pipe size, in.	Pipe length, ft	Fittings, ft	Total, ft
Header and riser runout....	2½	54	7	61
Riser....................	2	11	None	11
Radiator runout...........	1½	10	48	58

TABLE 3.6. FRICTION LOSSES

Pipe section	Pipe size, in.	Equiv. length, ft	Loss, oz 100 ft	Actual loss, oz
Header and riser runout...........	2½	61	0.67	0.41
Riser.........................	2	11	0.67	0.08
Radiator runout..................	1½	58	0.67	0.40
Total........................	...	..	....	0.89

Now we can set up the friction loss (Table 3.6). The total loss, being theoretically less than 1 oz, whereas we can handle 6.4 oz, suggests that we could reduce the size of our piping and save some money by so doing. However, the fact is somewhat different from the theory, and friction loss has a tendency to grow and grow as time roughens the surface of the pipe and stores the corners and fittings with corrosion and deposits of one kind or another. Those test figures in Table 3.1 are mute evidence of the dangers residing in a strict adherence to theory; a sufficient number of theoretically perfect installations gave enough trouble to demand research and action and safer guidance. Being wary designers who consider satisfactory operation of the system to be of utmost importance, we shall take heed of the experiences of others and let our pipe sizes stand.

Figure 3.8 illustrates an interesting fact in pressure calculations that many engineers, meeting the need for these calculations for the first time, do not immediately comprehend. It means that the pressure drop computed for equivalent length, pipe run *C*, is the pressure drop for the whole system. We do not add together the drops of all the individual runs, because, obviously, a system that is designed to handle the biggest drop of all will have no trouble with the smaller ones.

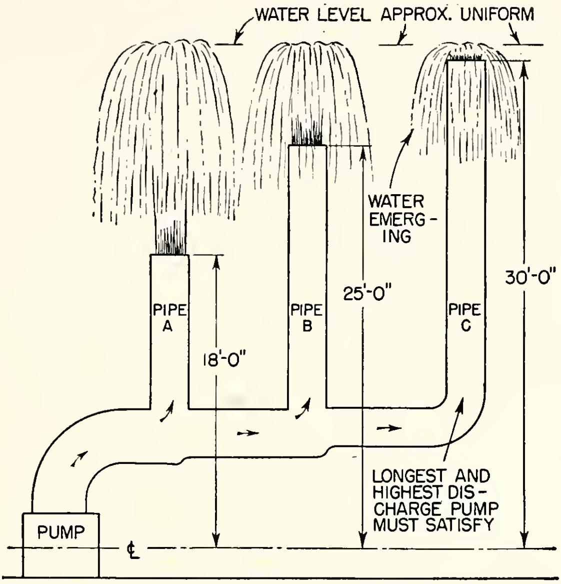

WATER LEVEL APPROX. UNIFORM
WATER EMERGING
PIPE A
PIPE B
PIPE C
18'-0"
25'-0"
30'-0"
LONGEST AND HIGHEST DISCHARGE PUMP MUST SATISFY
PUMP
℄

Fig. 3.8. Pump discharge pressure for various pipe lengths.

The same fact holds true with all fluids. For example, if we have a pump big enough to push oil from California to New York, the pump certainly can deliver a few gallons to St. Louis and Chicago in route.

Sizing the One-pipe Wet Return. We go about sizing the return piping and computing pressure drop exactly as we did with the steam piping, with one exception: we use a different table. For return piping, we use Table 3.2.

Since we designed the steam piping for an average drop of 0.89 oz per 100 ft, suppose that we accept the same maximum drop for our returns and see what we get.

Table 3.2 shows us that a wet return using 1-in. pipe will handle condensate from 500 sq ft of radiation. This is more than our 455 sq ft, and it appears ample. However, if there is one rule above all rules that the designer and contractor should respect, it is this: Never, never cut a job too close. There are so many variables of weather and human estimates of what constitutes heating comfort, all lying in wait to prove our design in error, that if we ignore foreseeable troubles like mill scale, saw blades, and other alien items left in the pipe, we are being downright foolhardy.

In view of these possible troubles, we will take the safe course and size our return not of 1-in. pipe, but rather of 1¼ in. The difference in cost is small, but the insurance large. This is a case in which judgment outweighs theory.

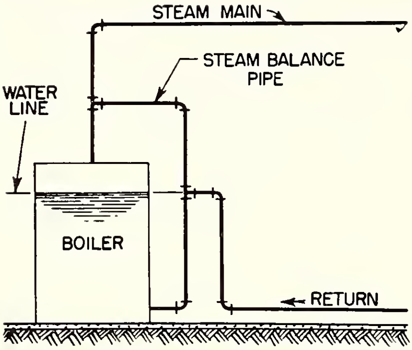


Fig. 3.9. The Hartford loop.

Sizing the One-pipe Dry Return. For a dry return, Table 3.2 shows a 1¼-in. pipe for our load. There is no doubt of this pipe being large enough under anything like normal conditions, but the average contractor will prefer to use 1½-in. pipe. This is because experienced contractors dislike having to come back to a job after it apparently is finished. Back calls cost time and money, and consequently present caution usually is the child of remembered grief.

The Hartford Connection. The Hartford connection may have originated in the offices of the Hartford Insurance Company, or perhaps was dreamed up by a man of the same name. Figure 3.9 offers a side view of the Hartford connection, and we observe that it is merely a pressure-balancing loop built up with a little pipe and a few fittings. By introducing full boiler pressure on the return side, the water in the boiler is prevented from leaving through the return piping.

Loss of boiler water by reversed circulation through the returns has occurred in the past, and it is likely to happen to us too if we neglect to provide a Hartford loop or, alternately, a check valve. If the check valve appears to be a simpler idea, we should reflect simultaneously that check valves sometimes fail to close tightly, especially if there is a little dirt on and around the seat. In fact, some engineers consider a check in this position to be a delusion and a snare, and will have nothing to do with such a scheme.

To be effective, the size of the Hartford loop piping must equal that of the return main. If large boilers are involved, increasing the size of the Hartford piping beyond the return pipe dimensions is desirable.

Another name for this return connection to the boiler is the "Underwriters loop."

Downfeed One-pipe Systems

Troubles of one kind or another that followed early installations of the one-pipe steam seem to have inspired various corrective treatments. The downfeed design illustrated by Fig. 3.10 may have originated from a desire to keep the steam and condensate always flowing in the same direction. Again, it may have been dreamed up as a method for circumventing the height limitation on conventional one-pipe risers.

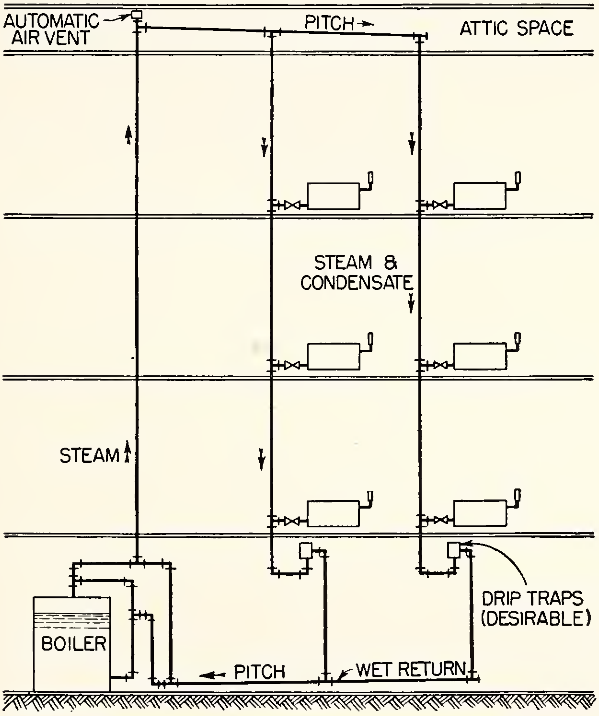


Fig. 3.10. Downfeed one-pipe system.

The downfeed system can be fairly costly to install for these reasons: First, we must erect a large, heavy, main riser from basement to attic and insulate it throughly. Second, we have the expense of distributing headers and branches in the attic, all of which also must be warmly overcoated with magnesia or asbestos insulation. Third, from the standpoint of building construction, there is need for a large pipe chase or, alternatively, the graceless pipe must stand out in the various rooms or halls where all can see, and disapprove. Few architects will accept the visible arrangement.

With the downfeed design, we provide the largest diameters for the downcomers at the top floor, for here we have the greatest volume of steam. Gradually then, the size of the piping may be decreased until, having served the last radiator on the lowest floor, we need pipe no larger than is required to carry away the condensate.

The employment of a trap at the base of each downcomer is desirable to prevent steam from entering the returns where steam performs no useful service.

Sizing Downfeed Piping. Tables 3.1 and 3.3 provide us with a working guide for sizing the downfeed systems. The big riser to the attic, with its steam and condensate flowing in opposite directions, must be sized for the full load. Each of the branches in the attic, being pitched in the direction of steam flow, will be sized to serve its particular load. The downcomers are sized also from the columns showing the steam and condensate flowing in the same direction.

Return piping follows the same system, the individual sizes being determined from Table 3.2 and then evaluated for any special conditions.

Two-pipe Systems

We now come to the design that provides two pipes for every radiator. One pipe supplies the steam; the other carries away the condensate. These connections are made to opposite sides of the radiator, so that the direction of fluid movement through the unit is always the same, whether steam or water.

The idea behind the two-pipe design is to avoid all conflict between steam and condensate.

In early two-pipe design, the thermostatic radiator trap was sometimes overlooked, or unavailable, and its normal position on the radiator outlet was usurped by a valve. Since this valve was left open necessarily so that condensate could escape from the radiator, steam escaped also, and many a fireman wondered why his heating system put out so little heat for so much coal.

Figure 3.11 illustrates a typical two-pipe system. The arrangement shown uses a receiver and pump to transfer the condensate back to the boiler. However, many two-pipe systems let the condensate flow back by gravity, entering the boiler through a Hartford loop, while others, generally of large size, employ a vacuum pump.

Use of a Vacuum Pump. Figure 3.1 illustrated a vacuum pump in wide use. The vacuum feature has several advantages. It permits us to eliminate air valves from the radiators and return piping, for all the air is collected and disposed of by the machine.

The vacuum pump enables us to heat our radiators with steam at subatmospheric pressures. We can do this because the flow of steam through a radiator depends on the relative pressures at the inlet and outlet. Thus, if we have 2 psig at the entrance and 1 psig at the outlet, steam will flow at a certain rate. The same tendency to flow occurs if the pressures are both below atmospheric: for example, if the pressure is 14 psia at the entrance and 12 psia at the outlet. We still have the 2-lb differential.

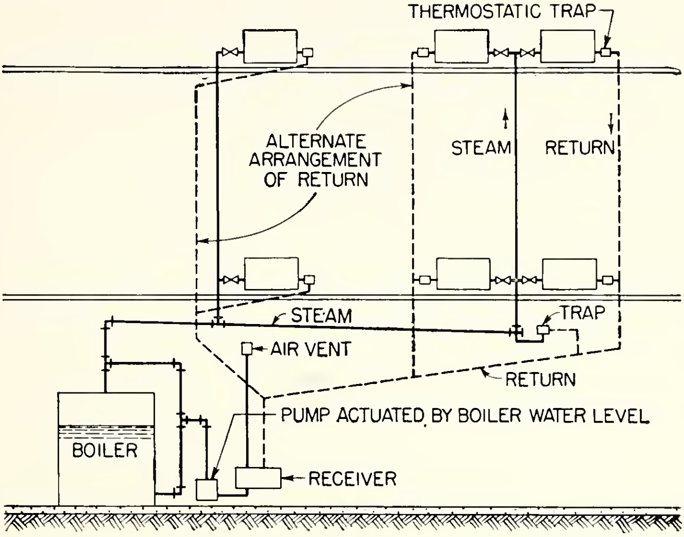

THERMOSTATIC TRAP
ALTERNATE ARRANGEMENT OF RETURN
STEAM
RETURN
STEAM
TRAP
AIR VENT
RETURN
PUMP ACTUATED BY BOILER WATER LEVEL
BOILER
RECEIVER

Fig. 3.11. Diagram of two-pipe system.

The advantage of subatmospheric pressures is most realized at the beginning and end of a heating season. At these times, when the need for heat is comparatively small, we can fill the radiators with vapor at 180 F (7.5 psia) instead of the 216 F steam at 16 psia (approximately 1.3 psig).

Obviously, we evaporate less water for our heating and thus save fuel. Here we see one of the advantages of the two-pipe system.

Lift Fittings. While we always endeavor to design our condensate returns to flow by gravity, there will be times and places that baffle our skill in achieving this end. The vacuum pump is our valuable ally in this situation, for with it we are able to use the lift fitting illustrated by Fig. 3.12. This ingenious arrangement of pipe and fittings helps mightily in getting our returns over a hump or up a hill. What happens is that we convert pressure differentials over the lift fitting into velocity, and this velocity enables the condensate to flow over the rise. The differential of course is developed by the vacuum pump.

Sizing the Piping for the Two-pipe System. Two-pipe systems can be laid out either on the upfeed plan with a header in the basement or as a downfeed arrangement with the supply headers in the attic.

In either case, we size the piping similarly to our sizing of the one-pipe systems, that is, we refer to Tables 3.1, 3.2, and 3.3 and take the information we need from the proper columns.

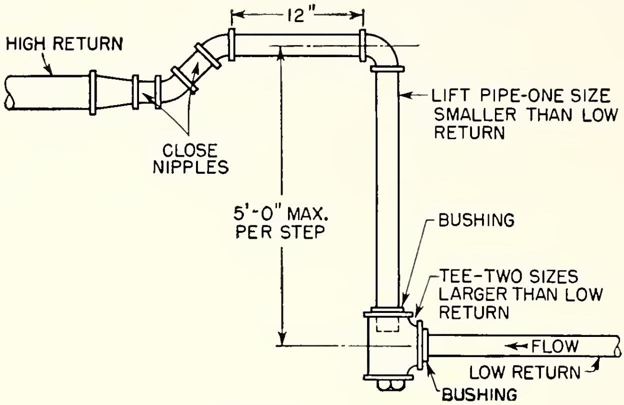

12"
HIGH RETURN
CLOSE NIPPLES
5'-0" MAX. PER STEP
LIFT PIPE-ONE SIZE SMALLER THAN LOW RETURN
BUSHING
TEE-TWO SIZES LARGER THAN LOW RETURN
FLOW
LOW RETURN
BUSHING

Fig. 3.12. Lift fitting.

Variations and Hybrids

Our basic designs for steam heating systems are only two in number: the one-pipe and two-pipe systems. Each of these is also subject to two forms of application; it must be either upfeed or downfeed.

This seems simple enough, but what original and adventurous thinkers in the heating field have done to modify the basic conceptions is sometimes incredible. One must see it to believe it. The following sections describe some of the more common arrangements that we shall meet (and perhaps use ourselves) from time to time.

Atmospheric Systems. The atmospheric system is open to atmosphere, hence its name. Figure 3.13 shows diagrammatically how such a system is laid out. On occasion the various vents tie into a single vent, and the writer has observed the functioning of such a system in a 15-story office building. In this case—and here is one of the weaknesses of the atmospheric design—more than atmospheric pressure became necessary to deliver heat to the upper floors. Consequently, the vent pipe frequently projected a plume of steam against the winter clouds, and since steam costs money to make, operations became quite expensive at certain times of the year. A back pressure valve ultimately was mounted on the vent but this was not altogether satisfactory either, since it trapped air in the system as well as steam. One thing led to another until eventually this system was no longer entitled to the "atmospheric" classification.

The delicacy of pressure balance in the atmospheric system persuades most engineers to avoid its use. Theory in this case is not easily realized in practice, the objective being to heat with steam that is not 1 deg hotter than necessary, and thereby save fuel.

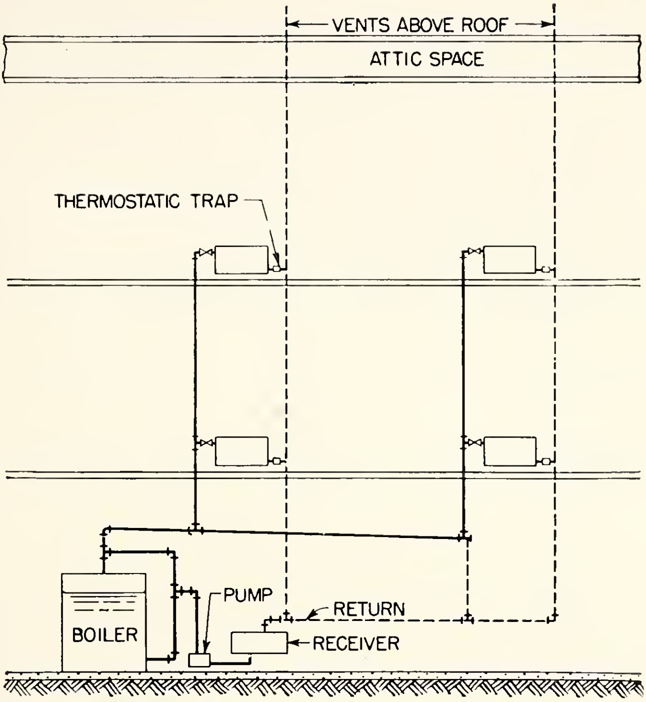


Fig. 3.13. Diagram of atmospheric system.

The vacuum pump and subatmospheric systems do a better job with less effort, and more surety.

Using Extra Drips. There is nothing fundamentally wrong with hybridizing a system, provided such hybridizing improves results. For example, if we have a one-pipe system with long basement headers, intermediate dripping of the headers can be a valuable precaution against faulty and sometimes noisy circulation.

Dripping individual risers also augments the heating efficiency of a system, but substantially adds to the cost. However, thermostatic traps on the bottom of the downcomers in a downfeed design can justify themselves very quickly in their saving of steam.

Again, if we have a building of various levels or stories, we might save money by using a one-pipe system in the lower wings, and two-pipe arrangements in the higher wings.

We will find opportunities to mix up radiators, convectors, warm air coils and blowers, and perhaps hot water heating, all for the good of the building under consideration. Our first thought is to provide satisfactory heating, and our second is to provide it at the lowest practical cost. If we have neither imagination nor ingenuity to do these things when needed, then we are not engineers.

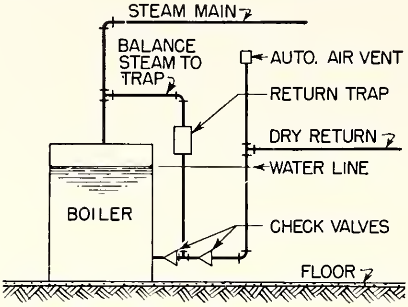


Fig. 3.14. Return trap installation.

Vapor Systems. Strictly speaking, vapor is steam at subatmospheric pressures. Nowadays there is a loose tendency to apply the term to the whole wide field of low-pressure heating, especially if the pressure varies from a few positive ounces to a few inches of vacuum.

The critical feature of a vapor system is observed in its low potential pressure of distribution. Let a little air creep into the piping, or a little rust increase the normal resistance to flow of valves and fittings, and we have insuperable blocks to vapor distribution.

Vapor systems release their air to atmosphere normally in the basement at a point well above the water line of the boiler, and as usual the vent is on the return. Sometimes if we are not careful, the boiler pressure may run up high enough to force a little water from the returns out through the vent; then we say with owlish wisdom, "We've blown the seal."

When this happens we have several possibilities of cure as follows:

1. We can reduce the pressure. This is the simplest thing to do, but since the cause of the high pressure in the first place is often the need for more steam in the radiators, cutting down the pressure may increase the complaints.

2. We can increase the size of the individual radiators. This is expensive, and hardly anybody will accept this cure.

3. We can install an automatic return trap. This is the most widely accepted method, and most contractors and designers will provide such a trap from the start.

Return Traps. Figure 3.14 shows how a return trap is tied into the system. We must start with a dry return, two check valves, a connection to the steam side of the boiler, and of course the trap itself. The trap contains a float which, when the water reaches a predetermined level in the trap, opens the steam valve, thereby equalizing the pressure on both the steam and water sides of the boiler. The water in the trap then flows readily into the boiler.

By use of the trap we can run up the pressure in the boiler to any degree we wish, for the water in the return never goes above the predetermined level, and we can never "blow our seal."

If the owner objects to the return trap on the grounds that it takes his system out of the strictly vapor class, then there is little we can do. Vapor has its limitations.

Some Tricks of Design

If genius is the capacity for taking infinite pains, then just as truly a satisfactory steam system is the result of taking a few pains in the design stage. The need to make corrections later can become costly in material, labor, and reputation.

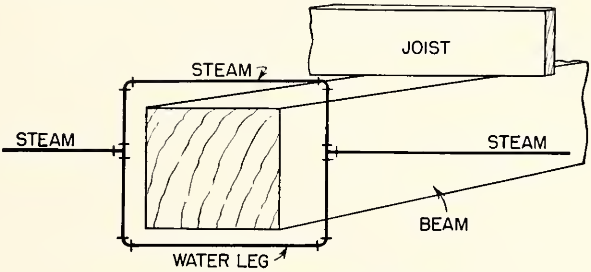


Fig. 3.15. Circumvention of a beam.

First Considerations. A first thought in any design is to make sure we can run our piping where and how we want to run it. An engineer who ran piping across windows and through columns and similar obstructions would cease quite soon to be an engineer. This thought is not elementary, for only the utmost care will prevent us from doing such things ourselves.

Nonetheless there will occur times in which such conflicts seem unavoidable; we simply have to set a column and a pipe in the same place. This is our opportunity to exercise resourcefulness and ingenuity.

Circumventing a Horizontal Beam. Figure 3.15 shows a method successfully used for getting a steam line around a horizontal beam. The steam pipe goes up and over; the water of condensation goes down and under. After the water bight under the beam is filled, water flows along the pipe in the normal manner. We thus avoid the creation of a water pocket which otherwise would obstruct the movements of the steam.

Piping in Walls and Floors. The space between floor joists, or between the studs of walls, sometimes can be used as passageways for piping. Many persons fear that putting hot pipes in these places invites fire. This worry is unnecessary, for actually the steam temperatures we use in heating practice are far, far below the ignition or charring temperatures of wood. In fact, close-fitting wood conduit has long been used around steam pipes for insulation and protection. Another method of insulating steam pipes is to pack them in wood shavings.

Radiators for One-pipe Steam. The valve of a one-pipe radiator can be quite easily a bottleneck in the passing of steam against the drainage of outflowing condensate.

A way to ameliorate the trouble is to be sure to install the radiator with a definite pitch toward the valve. After all, we know that a steam pipe requires a pitch of 1/4 in. for each 10 ft, while water of condensation needs 1/2 in. in the same distance.

As the distance from the inlet to the far end of a radiator may be from 3 to 4 ft, there is obvious need for grading of the unit. As a rule, lifting the far end 1/2 in. with a block under the legs is ample. Some radiators that heat poorly will perform quite satisfactorily after being given the pitch treatment.

Orifices. The use of orifices in radiator inlets or in piping branches is favored by many designers for obtaining a close balance of the system. The principle of the orifice is this: The discharge from an orifice remains constant so long as the absolute pressure on the low side does not exceed 58 per cent of the absolute pressure on the inlet side.

As an example of how this works out in practice, suppose that we consider the case of two radiators, say 150 ft apart. The inlet pressure at the first orifice is 2 psig; at the second, pipe friction has cut the pressure to 1.4 psig. Suppose, furthermore, that we desire each radiator to condense 100 lb of steam per hour.

Before we install orifices, we may discover the first radiator to be condensing far more than the 100 lb of steam to which it is entitled. It is, in short, overheating.

Now if we cut down pressure to bring the first radiator under control, the second suffers from starvation. Therefore what we do is to fit the first radiator with an orifice that will permit the entry of 100 lb of steam per hour while the pressure is at 2 lb, and while the outlet side is simultaneously at 58 per cent of 2 lb, or less. With this condition established, we can hold up the pressure to whatever value is needed for adequate heating of the second radiator, but the first one will continue to receive only the 100 lb just the same.

Control of this differential across the orifice may be realized through careful maintenance of the steam pressure or conversely by adjustments of the vacuum on the return side. In practice the maximum differential worked with is usually about 4 lb.

Orifices for this operation cannot be made satisfactorily from washers that we buy at the hardware store. The proper place to go for them is to the manufacturers of heating specialities, who calibrate them with care for the exact conditions that we specify.

Orifices are sensitive, and they are subject to corrosion and erosion. Use of the wrong material or size can easily make us worse off than we were in the first place, without them.

Water Hammer. If we hear a steam heating system emitting sounds suggestive of the anvil chorus, it is prime evidence that the steam and water within the system are in conflict.

Generally we consider two possibilities of cure. The first of these is to discover where water is accumulating and then to provide an extra drip, or drain, at that point. The second possibility is to improve the pitch of the piping.

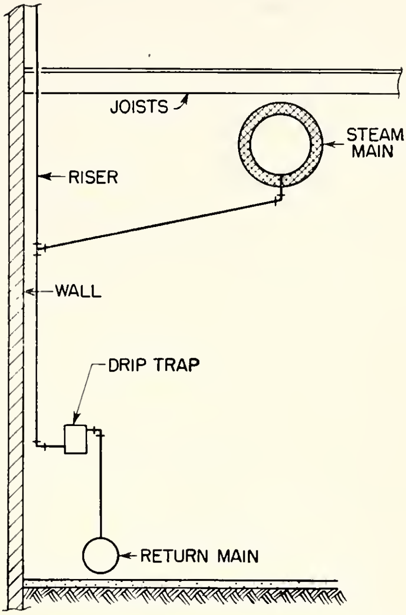


Fig. 3.16. Bottom take-off for risers.

The first cure is indicated on those systems where the attempt to hold down the cost has resulted in too few, or wrongly located, drip connections. The need for regrading indicates that somebody did not know precisely what he was doing in the first place.

Steam pipes should be pitched not less than $\frac{1}{4}$ in. for every 10 ft in the direction of steam flow; water piping $\frac{1}{2}$ in. per 10 ft. Unfortunately, providing these grades is not always easy, and in consequence headers may find themselves erected on the level, or even tilted in the wrong direction.

There is nothing to be done with the reversed pitch but to correct it, or at least bring it to the horizontal. For the horizontal pipe, we can perform a trick of design, illustrated by Fig. 3.16. Here we take the runouts from the bottom of the pipe, so that condensate forming in the headers soon finds itself smartly shuttled into the wet return.

Another way of avoiding the horizontal header problem is to switch our heating agent from steam to hot water. We can pump the hot water up hill and down dale to say nothing of the level. Many a hot water system has been preferred over steam for this very reason.

One thing we must remember when laying out a horizontal main, or branch, that is, to size it at least one step larger than we would select for an equivalent pipe properly pitched.

Questions

1. Why is the movement of steam impeded by the presence of water or air?
2. Why does saturated steam provide a better heat emission than superheated steam.
3. What are the advantages of the one-pipe system?
4. What are the advantages of the two-pipe system?
5. You are called upon to design a steam heating system for a bungalow. You can locate the boiler in the basement. Would you design for one-pipe or two-pipe installation? Why?
6. Why is a one-pipe system impractical for houses without basements?
7. What are the advantages of gravity return? The disadvantages?
8. Calculate the clearance between the low end of a steam main and the boiler water line, if the pressure differential is 22 oz.
9. What size steam header would you use for 1,500 sq ft of radiation and 4 oz pressure drop per 100 ft of equivalent pipe length?
10. A riser runout serves 300 sq ft of radiation. What size should it be if it is 6 ft long? 12 ft long? How much pitch would you give it?
11. How does a vapor system differ from a pressure system?
12. Referring to the steam table, what is the difference in volume between steam at 5 psig pressure and vapor at 170 F?
13. A steam main is 350 ft long. What is the minimum difference of level you must provide between the beginning and end of this main to provide adequate drainage?
14. What is the function of a return trap?

CHAPTER 4

HEATING BOILERS AND STEAM GENERATORS

Heating Boilers, General

A heating boiler is a boiler designed to operate at pressures not exceeding 15 psig for steam and 30 psig for water, and is built primarily to generate steam or hot water for comfort heating.

Materials of Construction. Cast iron is widely used in the construction of these boilers, particularly in the small and medium sizes. Steel is found more often in the large boilers.

However, there is no sharp dividing line in the choice of metals. Builders' catalogues show cast-iron boilers with ratings up to 15,000 sq ft of connected radiation, and steel boilers for as little as 275 sq ft. We can buy pretty much what we want.

Historically we might observe that boilers have been built of wood, iron stays being used to hold them together against the internal pressure, like hoops around a barrel. Nonferrous metals have been used also, but since these materials are more expensive than cast iron and steel, their popularity has languished. In the future we probably shall see boilers formed of plastic, just as soon as somebody finds a way to ensure the integrity of the plastic at the temperature of boiling water.

Boiler builders issue their specifications itemizing the materials of construction, and since these may vary considerably in the same boiler for the different parts, we should acquaint ourselves with the nature of these metals (see the data in Table 4.1).

Cast-iron Heating Boilers

Ratings of Heating Boilers, Cast Iron. Builders' catalogues show heating boilers rated generally in terms of connected radiation. Sometimes a boiler so rated may be designed with some inherent overload capacity in order to meet pickup demands and to compensate for piping losses, or it may not be so designed. To standardize ratings therefore, the Institute of Boiler and Radiator Manufacturers has set up a rating procedure as follows:

The "net IBR rating, square feet" shall mean a load based on the actual installed radiation, *plus* an inherent capacity allowance provided by the builder for meeting the pickup and piping loss demands. Thus an engineer or contractor, having calculated the standing radiation for any given job, need not worry his head about adding something for pickup and piping losses if he selects a boiler on the net IBR basis. The excess capacity is already there.

Table 4.1. Description of Ferrous Materials*

Metal	*Description*
Wrought iron	Contains a measure of slag, usually less than 2% which encases the grains of the metal and resists corrosion. Carbon content is less than 0.10% in the good grades
Charcoal iron	Produced in a blast furnace using carbon fuel. Very pure wrought iron is sometimes called charcoal iron
Pig iron	Produced in a blast furnace. Contains from 2 to 5% carbon. When made with carbon fuel, it is termed charcoal pig iron
Malleable iron	Another name for wrought iron. Is not brittle and is readily worked
Malleable cast iron	When cast iron has been refined by annealing in oxides, it becomes malleable iron
Cast iron	Pig iron poured into molds to produce iron shapes of the size and form desired by industry. Has high compressive strength
Gray iron	Gray iron is cast iron with an inclusion of graphite. The graphite gives the casting toughness. When a gray iron casting is broken, the surfaces of the fracture show a characteristic gray color, hence the name
Steel	Iron that contains no more than a trace of slag, or no slag, and in which the carbon is generally less than 2½ per cent, may be termed steel
Semisteel	Iron castings which contain 25% or more of steel are made by melting steel scrap with the pig iron, and are called semisteel. The steel produces castings with a finer grain and greater strength than is found normally in cast iron

* *Note:* This table does not attempt to enumerate the physical properties of the metals listed, since these vary widely as the proportions of the ingredients vary.

The term "gross IBR output, Btu per hour" means the actual total capacity of the boiler, as proved by the builder on test. The difference between the "net" and "gross" ratings, therefore, is the amount of excess capacity available for pickup and piping.

We should note from the start that a square foot of steam radiation under the IBR Code is based on a heat emission of 240 Btu per sq ft of radiation per hour. Thus, 100 sq ft of radiation will put out 24,000 Btu per hr.

Hot water radiation, with natural circulation as the basis of the computation, is considered to put out 150 Btu per sq ft of radiation per hour. The average temperature of the hot water in the radiator at this rate of emission is 170 F.

When we circulate hot water with a pump, we can run the temperature up to 215 F, and in consequence the heat emission from the hot water radiators increases substantially. In practice, this means that we can reduce the amount of hot water radiation needed simply by increasing the temperature of the water. On the other hand, if the hot water drops below 170 F, the heat emission will be less in proportion, and to meet any given load we must provide more radiation.

Table 4.2 sets up the correction factors to be applied to radiation calculated on the 150 Btu emission basis, in the event we are going to circulate water at temperatures either above or below the basic 170 F. As an example of how this works out, suppose that we must provide hot water radiation for a new house having a calculated heating load of 150,000 Btu per hr. With 170 F water and a heat emission of 150 Btu per sq ft, the radiation required will be 150,000/150 = 1,000 sq ft.

However, we will heat up the water to say 195 F, and by referring to Table 4.2, we discover the correction factor to be 0.75. Multiplying 1,000 by 0.75 gives us only 750 sq ft.

The importance of this reduction in radiation, if we must pay say 70 cents per square foot for the radiation we buy, is obvious. Two-hundred fifty square feet at 70 cents per square foot is $175. This saving is one reason for the growing preference of designers for circulating hot water by a pump.

TABLE 4.2. CORRECTION FACTORS FOR HOT WATER TEMPERATURES

Avg. temp. of water in radiator, F	Emission, Btu/hr/sq ft	Correction factor
150	110	1.363
160	130	1.153
170	150	1.000
180	170	0.938
190	190	0.790
195	200	0.750
200	210	0.701
210	230	0.652
215	240	0.625

Heat emissions for both steam and hot water are considered to be taking place with a room temperature of 70 F. Actually, if the room's temperature is either higher or lower than 70 F, emission from radiator surfaces will be either less or greater than the standard, because the velocity of heat loss from a hot surface to the surrounding air is affected by the temperature difference between them. Thus if room temperature is higher than 70 F, the differential is smaller and of course the emission is less. Conversely, if the room is at say 60 F. the temperature differential is greater and the radiator will lose its heat at a faster rate.

For laboratory work, a knowledge of the exact rate of emission may be necessary, but for ordinary heating calculations we can figure that our heat emissions are taking place at 70 F. So many other variables affect the proper heating of a room that any attempt to calculate emissions to the ultimate Btu is simply a waste of time. For example, a window sash may warp and let in more cold leakage than we had expected, and what that could do to a precise calculation might be quite painful.

Table 4.2 can be used to determine the amount of radiation necessary, directly from the Btu load. Thus, 150,000 Btu/200 Btu = 750 sq ft which should be provided for water at 195 F.

However, at all times we must remember that the IBR boiler ratings, as they appear in the manufacturers' catalogues, are based on an emission of 150 Btu with 170 F water. This means we must convert all our radiation to the IBR basis when selecting a boiler from a catalogue. Thus, if we use 195 F water requiring only 750 sq ft of radiation, the proper boiler is *not* one rated in the catalogue at 750 sq ft, but one rated at 1,000 sq ft. The reason for this becomes clear if we express the radiation in Btu instead of square feet; 750 sq ft × 200 = 150,000 Btu, whereas 750 × 150 (the IBR basis) is only 112,500 Btu. Any boiler in the catalogue rated at less than 1,000 sq ft would be too small.

If we are heating with steam, no alteration of steam temperatures need enter the calculation. The radiation is based on a heat emission of 240 Bru/hr/ft^2. Thus our house load of 150,000 Btu would require 150,000/240 = 625 sq ft. To select a boiler for this job, we merely refer to the catalogue and find one that is rated for at least 625 sq ft of direct steam radiation.

Forms of Cast-iron Boilers. Cast-iron heating boilers frequently are built up of sections, as illustrated by Fig. 4.1. These sections are produced in the foundry as unit castings, and the boiler is then formed by assembling together as many of these castings as the manufacturer considers necessary to satisfy the rating.

Sectional boilers have several important advantages. For instance, the standardization of the sections keeps down the over-all cost. A boiler manufacturer can stock these sections and then, to fill an order, take as many from the storeroom as needed. This way of doing business permits the manufacturer to take advantage of fluctuations in the costs of labor and material.

Another advantage has to do with shipping and erecting on the job. For instance, a comparatively small truck can deliver a very large boiler, simply by handling one or more sections at a time. At the site, access to the boiler room, and particularly existing boiler rooms, may be quite restricted. In these cases, a boiler that cannot be taken in piecemeal, may not go in at all, unless we cut out part of the wall.

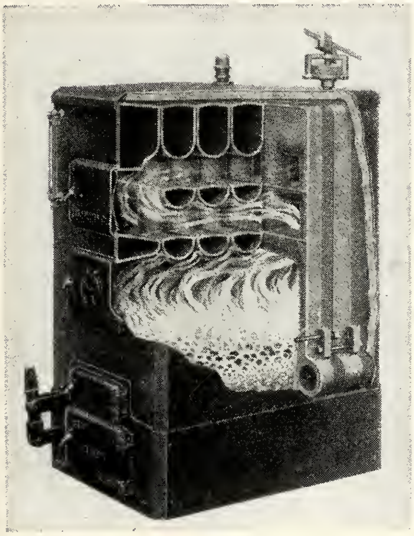


Fig. 4.1. Cutaway view of hand-fired cast-iron sectional boiler. *(Courtesy of Burnham Boiler Corporation.)*

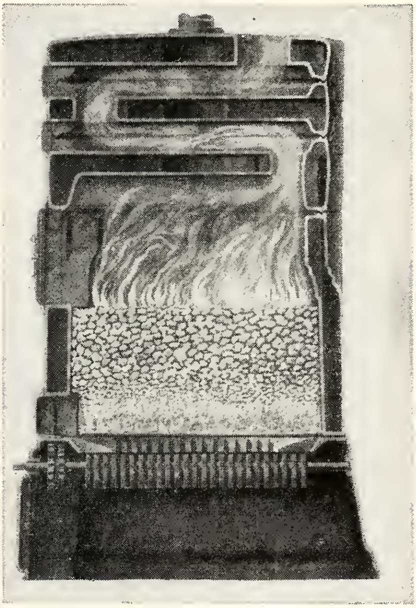


Fig. 4.2. Cutaway view of coal-fired round cast-iron boiler. *(Courtesy of Burnham Boiler Corporation.)*

Sectional boilers as a rule will pass readily through any standard doorway.

Figure 4.1 shows a sectional cast-iron boiler set up for operation, with some of the external surfaces cut away to permit our viewing the internal arrangement of the sections.

Figure 4.2 illustrates the round or cylindrical form of the cast-iron boiler. The sections of this type sit one on top of the other, like layers in a cake.

Figure 4.3 shows a cast-iron boiler with what is known as a "wet base." The advantage of the wet base is in providing a furnace completely surrounded by water. We can set a wet-base boiler on a wooden floor, and generally even the fire marshall will accept it in that position.

All cast-iron boilers, whether sectional, round, wet base, or of any other special characteristic, may be fitted in a steel cabinet of pleasing appearance. This cabinet conceals the insulation, the oil burner or gas burner, and provides a neat, clean, and much improved appearance for most basements and boiler rooms.

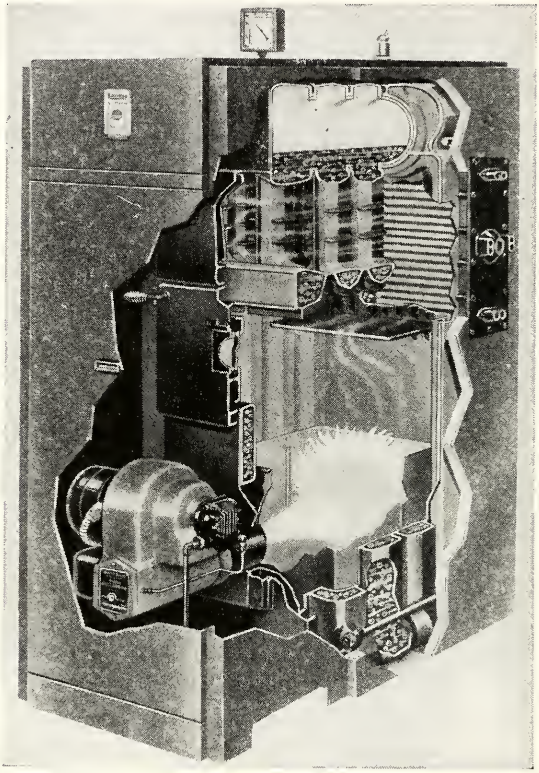


Fig. 4.3. Cutaway view of oil-fired wet-base cast-iron sectional boiler. *(Courtesy of American Radiator and Sanitary Corporation.)*

The cabinet adds to the price of the boiler, but to most buyers the extra cost seems justified.

Cast-iron boilers may be fired with any of the conventional fuels, such as coal, oil, or gas. In the larger sizes, coal is generally fired with a mechanical stoker. The stoker tends to be too big for concealment in a cabinet, consequently some additional floor area may be required in these cases.

Steel Heating Boilers

Steel boilers are offered in a considerable number of sizes, shapes, and types. Some are round or cylindrical; some are designed for vertical setting; some for horizontal setting. Some of the designs are very clearly differentiated in their forms, for instance, we have the horizontal return tubular boiler, the locomotive or firebox boiler, and the Scotch boiler. Each of these units has its own peculiar characteristics, not the least important being certain suitabilities for any given space.

Rating of Heating Boilers, Steel. The Institute of Boiler and Radiator Manufacturers, which deals with the cast-iron units, has its counterpart in the steel boiler field, namely, the Steel Boiler Institute, Inc.

The "net SBI Rating in square feet of radiation" means the actually connected load of radiation, *plus* an allowance built into the boiler by the manufacturer to meet the demands of domestic hot water. Notably, the net SBI rating *does not include* any capacity for meeting piping losses as such, or for load pickup.

The piping losses, however, may be neglected if they are not estimated to be in excess of 20 per cent of the other total connected load, since there is normally enough reserve capacity to absorb these losses. The pickup is another matter, and we should add some percentage to our calculated load to meet the demands of a cold system on starting up. Steam in cold pipes and cold radiators melts like snow shoveled into a furnace.

The pickup is a matter of judgment. In most cases we shall be safe if we add from 25 to 30 per cent of the load imposed by the connected radiation.

Steel boilers can be and are used with hot water systems. To select a boiler for hot water, we need only follow the same procedure that we worked out under the IBR ratings for cast-iron boilers. That is, we must convert our calculated square feet of radiation to the equivalent Btu per hr emission basis, and study the SBI ratings to find a boiler on that basis.

Sometimes we shall find a boiler which is quoted, not on the "net SBI" basis, but simply as "SBI rating." The omission of the word "net" is quite important. Without it, the "SBI rating" means the gross rating of the boiler. The gross rating includes all loads specifically found under the net rating *plus* a calculated allowance for piping losses. The gross rating when we find it, however, does not relieve us from the need to add something for pickup.

The SBI rating has certain other implications. For instance, the Steel Boiler Institute distinguishes between boilers used for commercial purposes and for residential service. In each classification, minimum design requirements must be met or the SBI label may not be mounted on the boiler.

Let us examine some of these requirements. Take the relationship between boiler heating surface and connected load. In the commercial class, a hand-fired boiler must provide a minimum of 1 sq ft of heating surface for every 14 sq ft of equivalent direct radiation, and this direct radiation must be representative of the entire load, pickup and all. For mechanically fired boilers, the ratio goes up to 1 to 17.

For residences, the ratio is still 1 to 14 for hand-fired boilers, but the equivalent direct radiation is calculated on the *net* SBI basis, rather than on the total load. This relaxation of the requirement is a recognition of the less demanding nature of residential heating. The same relaxation applies to mechanically fired residential boilers, although the 1 to 17 ratio is maintained.

Other important standards for residential boilers are as follows:

1. If oil-fired, the burner must be set to produce at least 10 per cent CO_2 in the flue gas.
2. The flue-gas temperature shall not exceed 600 F when the boiler is operating at 150 per cent of the *net* SBI rating. Also the efficiency shall be at least 70 per cent under this degree of overload.

Hand-fired Grates. For hand-fired residential boilers, the SBI code demands a grate area calculated in accordance with the following formula: From the square root of a quantity which is one-ninth of the net SBI rating plus 600, subtract 8, and the result is the smallest grate area acceptable.

How this works out is demonstrated best by example. Suppose that we are interested in a hand-fired boiler which has a net SBI rating of 600 sq ft. The grate needed then is found this way:

$$\text{Grate area, sq ft} = \sqrt{\frac{600 + 600}{9}} - 8 = 3.6 \text{ sq ft}$$

Steam Boiler Institute Furnace Volumes. The Steam Boiler Institute Code stipulates certain minimum furnace volumes, a most important factor in obtaining good combustion.

In commercial boilers, the furnace must be designed with at least 1 cu ft of volume for every 140 sq ft of total equivalent direct radiation. The SBI rating may be used as the total load.

For residential boilers, the ratio is 1 cu ft of volume for every 110 sq ft of steam radiation, based on the *net* SBI rating. The boiler in each case is considered to be mechanically fired.

Round Steel Boilers. Figure 4.4 illustrates one of the small steel boilers of what we might call the round or cylindrical design. This interesting unit comes in small sizes, from 275 to 570 sq ft of steam, net SBI rating.

The fire tubes are seen to be fitted with what the builder calls "spinner blades." The purpose of these twisted ribbons of steel is to give the gases of combustion a whirling motion and thereby increase the rate of heat transfer. Other builders may use the term "retarders" for approximately the same thing. There is a difference of opinion among engineers as to the real value of these ribbons. Certainly they increase the draft loss through the boiler, a condition that grows worse as soot collects with use. The writer has been obliged to remove them in some installations to obtain the full rating of the boiler.

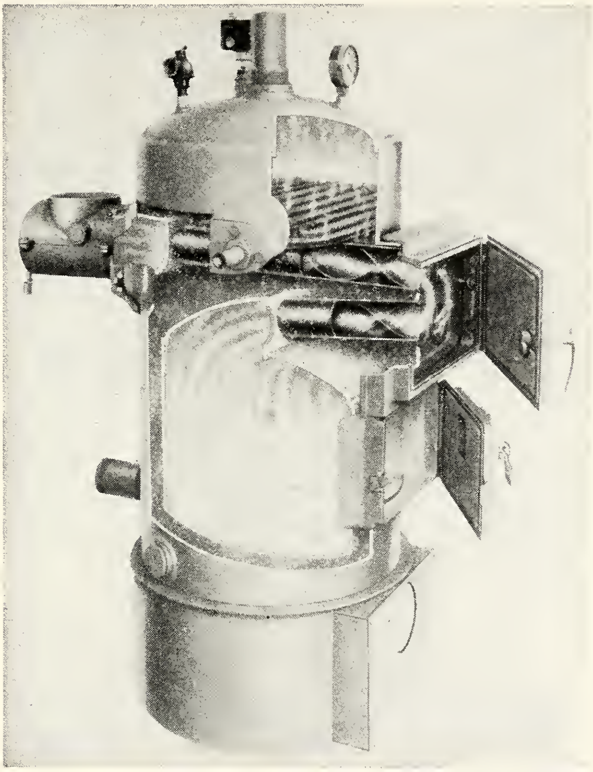


Fig. 4.4. Cutaway view of round boiler, showing domestic water heating coil set horizontally in dome section. Note gas twisters in flue-gas passages. (*Courtesy of Kewanee Boiler Corporation.*)

The built-in coil for heating up domestic hot water is shown in the top of the boiler.

This boiler is readily converted from hand to mechanical firing or vice versa simply by changing the base.

Figure 4.5 shows us another type of round boiler, designed primarily for hot water heating systems. The mounting of the burner on top of the boiler economizes on floor space, enabling installations in very close quarters. This boiler is rated up to 2,000 equivalent square feet of steam radiation, and 3,200 of water. Obviously these ratings are on the standard basis, as the ratio of 2,000 to 3,200 is the same as 150 Btu per sq ft to 240 Btu, the standard emissions for steam and hot water radiation.

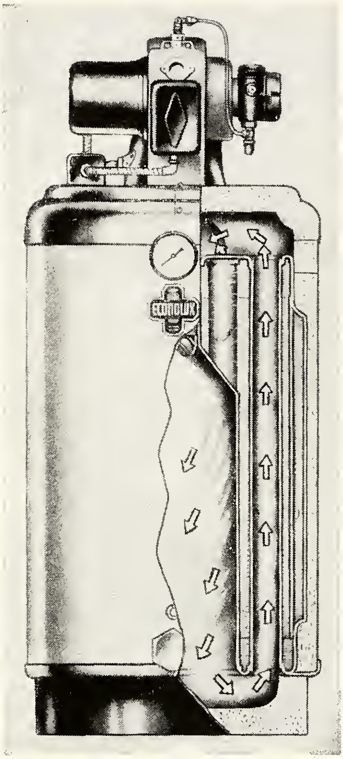


Fig. 4.5. Vertical type of boiler for hot water heating systems. (Courtesy of S. T. Johnson Company.)

Scotch Boilers. Figure 4.6 illustrates this interesting and ancient form of the steel boiler. The cylindrical furnace is completely surrounded by water, and the gases of combustion after leaving the furnace for a vertical combustion chamber return parallel to the furnace through boiler tubes which are also surrounded by water.

The cylindrical furnace is strengthened by its circumferential corrugations, a necessary feature in view of the water and steam pressures to which it is subjected. The corrugations also increase the hot surface in contact with the water.

Scotch boilers may be fired either by hand or mechanically, using any available fuel. When firing with oil, the furnace may be lined with refractory brick for a portion of its length in order to prevent the direct impingement of the flames on the metal corrugations.

The capacity of Scotch boilers can be increased, not only in over-all size, but in the number of furnaces. Four furnaces in a medium-sized boiler are not uncommon. If the load falls off, the fires are pulled from one or more of the furnaces, thus providing good flexibility of operation.

Just to avoid being surprised, we should note that Scotch boilers are built at times with furnaces at both ends. In these designs the combustion chambers are set back to back in the middle of the boiler. Such boilers are termed "double-ended boilers."

Some builders provide a small, extra drum on top of the Scotch boiler to facilitate the delivery of dry steam. This drum is called a separating drum, or, frequently, a dry drum.

Operators of Scotch boilers must be careful to avoid overheating of the furnaces. The furnace steel tends to weaken when overheated and may collapse from the boiler pressure. A frequent check of the trueness of the furnace, called "tramming," should be a matter of regular maintenance. Any evidence of the furnace being out of round may be indicative of a past overheating and potential weakness in the metal.

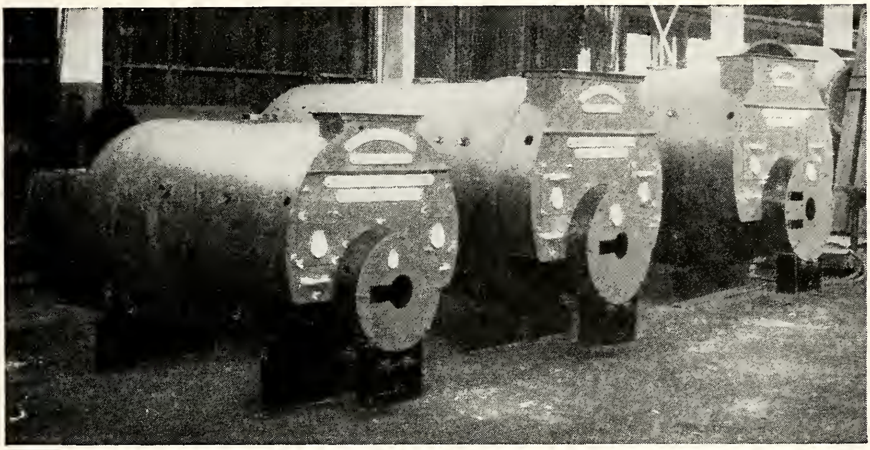


Fig. 4.6. Group of single furnace Scotch boilers waiting shipment. (*Courtesy of the Bigelow Company.*)

Return Tubular Steel Boilers. In Figs. 4.7, 4.8*a*, and 4.8*b* we find a form of boiler sometimes used for heating installations, but more often engaged in supplying steam at higher pressures to steam engines and industrial processes.

The horizontal return tubular boiler does not contain an integral furnace, which must be built separately of brick and refractory beneath it. The bottom of the boiler between the front of the furnace and the bridge wall is exposed to the direct heat of the fire. The gases of combustion pass over the bridge wall and then reverse their direction of flow and return to the front of the boiler through the tubes which are submerged in the water space of the boiler.

Stacks for return tubular boilers in the small sizes frequently are mounted directly on the boiler and then steadied by guy wires fanned out to ground anchors.

Horizontal return tubular boilers are simple in design, inexpensive to buy, easily moved from place to place. If we choose one for our heating system, we must include in our cost budget the brickwork of the furnace and the suspension steel.

Obviously this position of the boiler on top of the furnace demands a certain amount of headroom, more than is needed for a Scotch boiler or a comparable size of cast-iron sectional boiler.

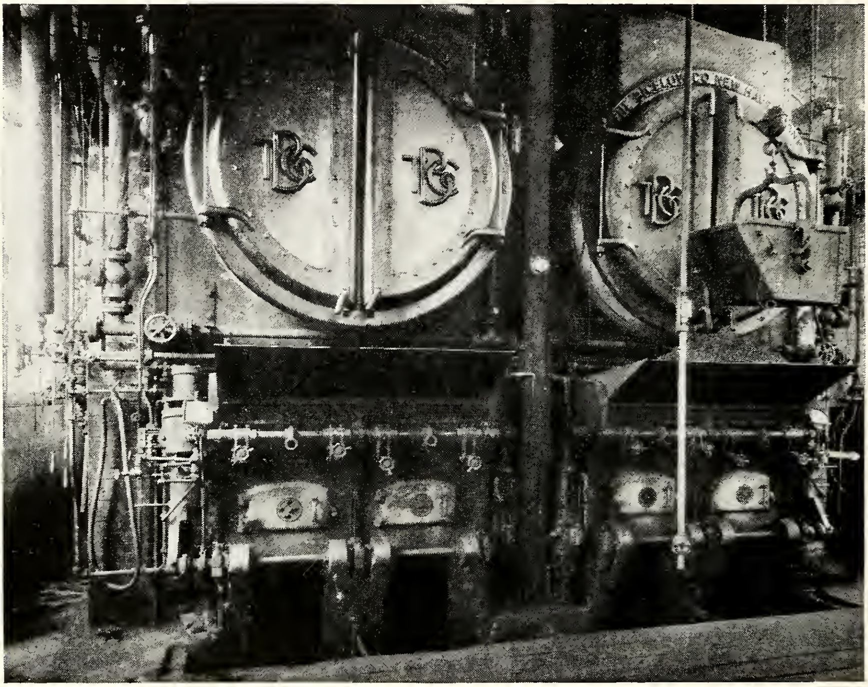


Fig. 4.7. Front view of stoker-fired horizontal return tubular boilers set in battery. Note coal supply equipment and ash removal alleys under boilers. (*Courtesy of the Bigelow Company.*)

Steel Firebox Boilers. In Fig. 4.9 we observe a typical boiler of this kind. There is some resemblence to both the Scotch and return tubular boilers in the tube arrangement. The furnace is underneath the fore part of the boiler as with the return tubular boiler, but in this firebox design the furnace is an integral member of the boiler, and is supplied by the builder. The sides of the furnace are surrounded by water, as is the vertical combustion chamber, thereby increasing the heat transfer surface.

The boiler shown in the illustration is called a "two-pass" boiler, which means that the gases travel twice through the water-surrounded tubes on their way from furnace to stack.

Occasionally corrugations are pressed into the steel sides and top of the furnace for the same purpose that they are used in the Scotch boilers, that is, for greater strength and increased heat transfer surface.

When these boilers are fired by oil, a refractory furnace, or well, must be built in the base to contain the flame. This is because the impingement of the flame on bare metal, when driven by the pressure of the nozzle

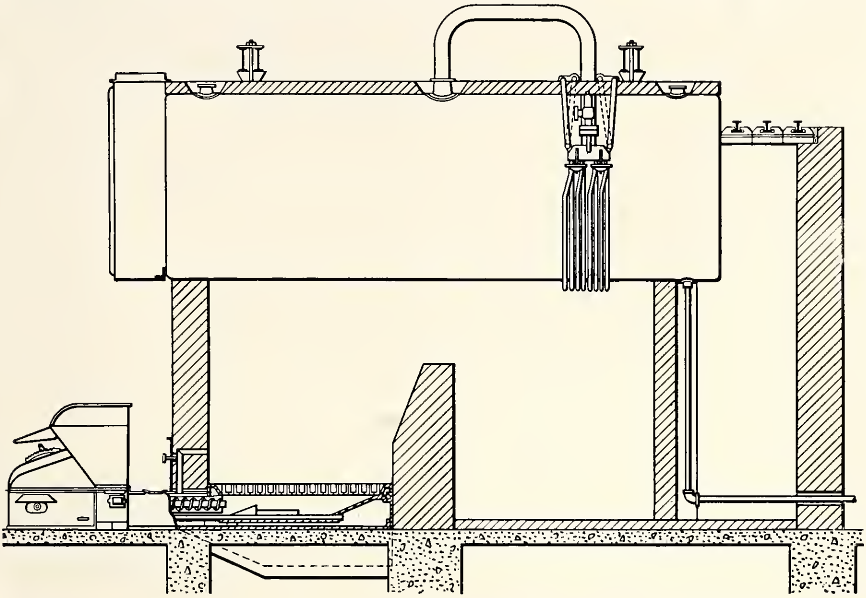


Fig. 4.8*a*. Side elevation of horizontal return tubular boiler. Note strap-type superheater. (*Courtesy of Combustion Engineering-Superheater, Inc.*)

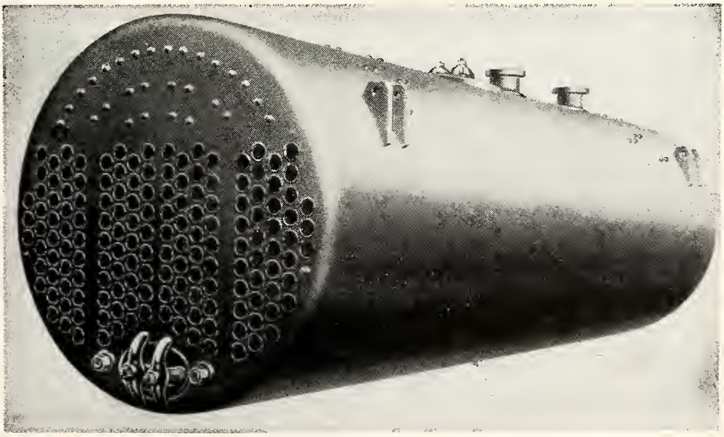


Fig. 4.8*b*. HRT boiler ready for shipment. (*Courtesy of Combustion-Superheater, Inc.*)

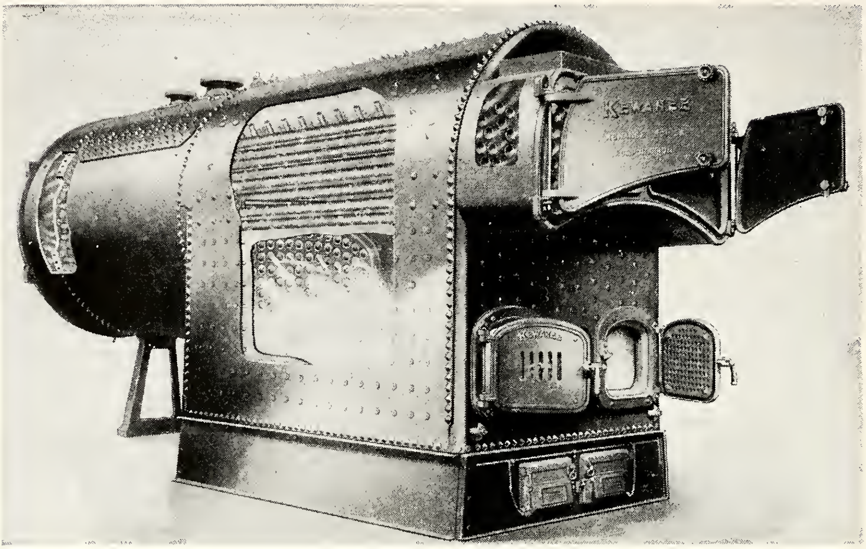


Fig. 4.9. Locomotive or firebox type of boiler. (*Courtesy of Kewanee Boiler Corporation.*)

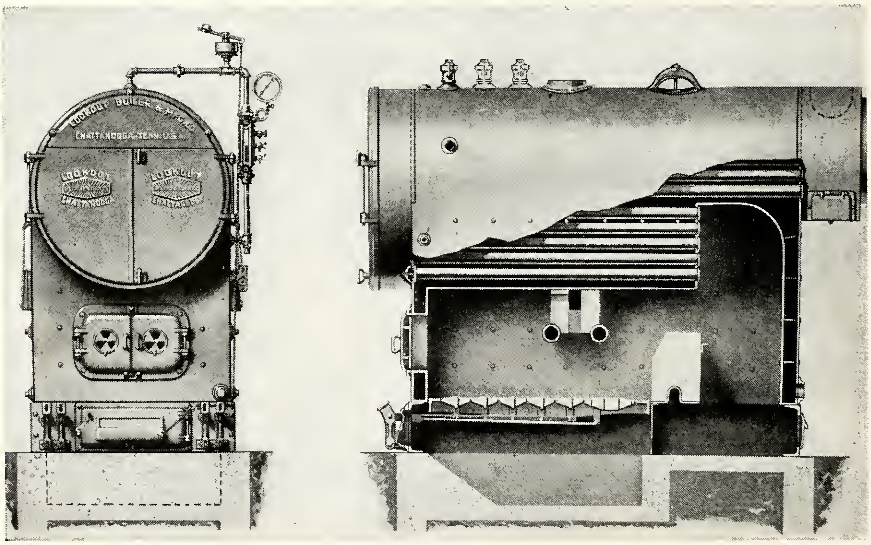


Fig. 4.10. Boiler with smokeless arch. (*Courtesy of Lookout Boiler Company.*)

blast, tends to have a blowtorch action which is destructive to the metal. Also, the comparatively cold surface of the steel checks combustion and thus causes smoke and soot.

Refractory brick linings on built-in furnaces are also necessary for stoker firing. Here again we have high-temperature fires whipped vigorously by the forced draft fan.

At times, the steel base is omitted and a brick foundation is furnished instead. This design is common for oil- and stoker-fired units, being a natural development of the essential, refractory furnace.

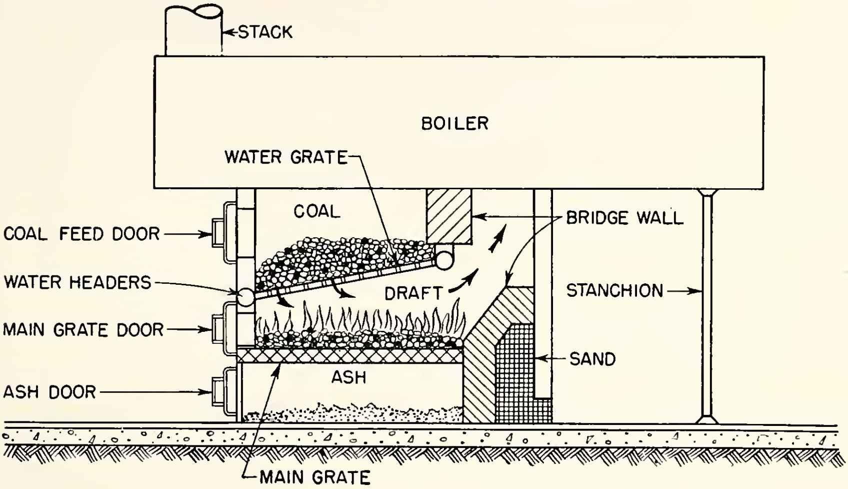


Fig. 4.11. The Downdraft Furnace.

The Smokeless Arch. High volatile bituminous coal is frequently a smoking coal unless the furnace designer takes some special precautions to guard against smoke. One device is termed the "smokeless arch," shown in Fig. 4.10. The function of the smokeless arch is to introduce hot refractory surfaces in the stream of combustible gases leaving the firebed, thereby igniting them. At the same time and place, some air is injected into the gas stream which furnishes oxygen needed for sustaining combustion.

The Downdraft Furnace. Figure 4.11 illustrates another plan for eliminating, or at least reducing smoke. Green coal is fed to the furnace on what is called a water grate, the tubes of the grate being part of the water system of the boiler. The water keeps the metal from burning up in its extremely rigorous position over the fire. The water grate terminates against an overhead refractory arch.

In operation, combustible gases are driven off the green coal in the water grate by the heat of the furnace. These gases, which are the main components of smoke, find their only paths to escape leading downward through the water grate, thus proceeding to the fire, where they are consumed.

The coal on the water grates eventually falls through to the main grate where combustion is completed.

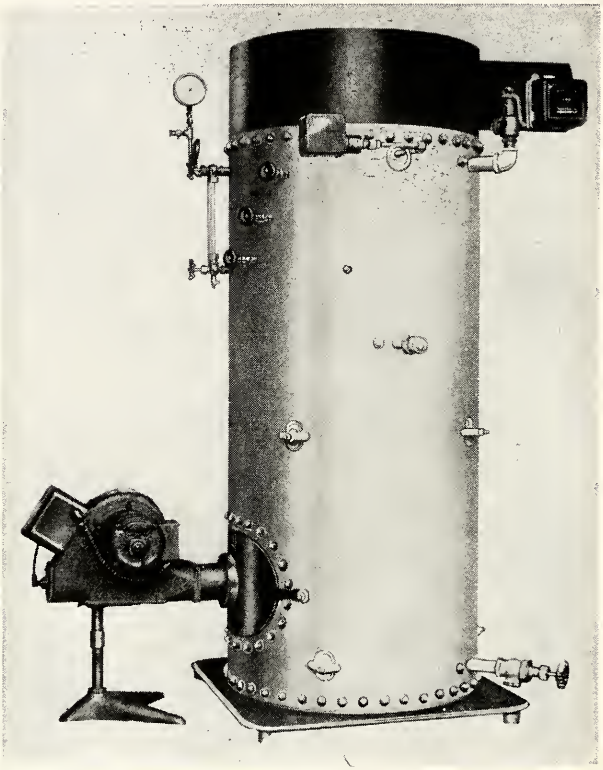


Fig. 4.12. Oil-fired vertical boiler. *(Courtesy of Lattner Manufacturing Co.)*

Some care must be exercised in a choice of coal for a downdraft furnace of this design. Caking coals, for example, will tend to build a hard crust over the water grate and thereby obstruct both the downward passage of the gases and the ultimate falling through of the burning coal to the main grate.

Reversed Arrangement of Firebox Boilers. The illustration in Fig. 4.9 shows a firebox boiler with the smoke pipe in the front. In this design, the tubes of the first pass leave the combustion chamber from the rear tube sheet, whereas in Fig. 4.10 we observe by contrast that the tubes leave the front sheet of the combustion chamber.

The significance of this contrast is important at times when we must install boilers in existing buildings, or in new buildings which have certain space limitations, or arrangements. For instance, the location of the chimney is frequently an architectural problem that is solved only by handing arrangement difficulties to the heating system designer. At these times, it is a great comfort to know that we can obtain a boiler with the smoke outlet at either the front or the rear.

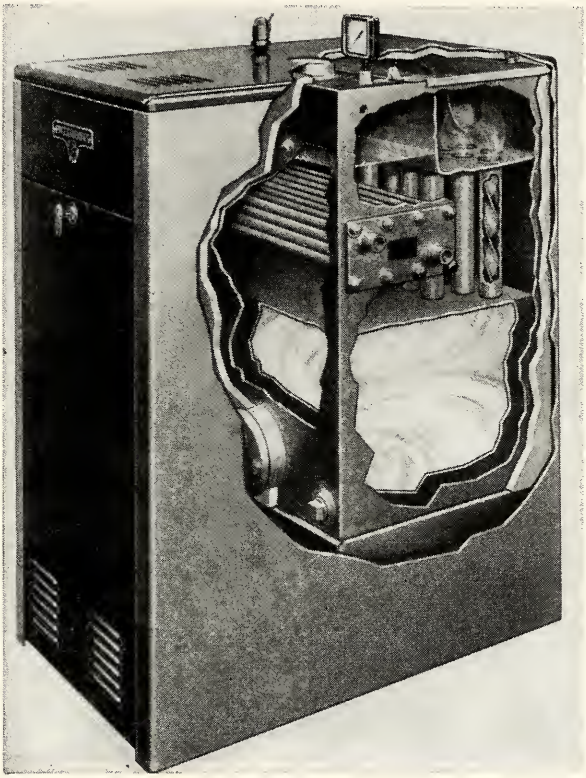


Fig. 4.13. Residential boiler with vertical fire tubes. Note horizontal domestic water heating coil. *(Courtesy of Fitzgibbon Boiler Company, Inc.)*

Many of the firebox boilers can be obtained in knocked-down sections, a definite asset in view of shipping and handling difficulties in many installations.

Vertical Steel Boilers. Figure 4.12 shows a boiler set on end; it is suitable for both heating and process steam demands. This boiler is a single-pass unit, the gases from the furnace going directly upward through the tubes and to the stack. This design certainly conserves floor space, but its demand for head room is not always acceptable. Efficiency is low, since the opportunity for heat exchange during the gas travel to the stack is short.

Another type of vertical boiler, recently developed, is illustrated by Fig. 4.13. This design is intended for home and similar small installations, with rating between 500 and 3,000 sq ft of equivalent direct steam radiation.

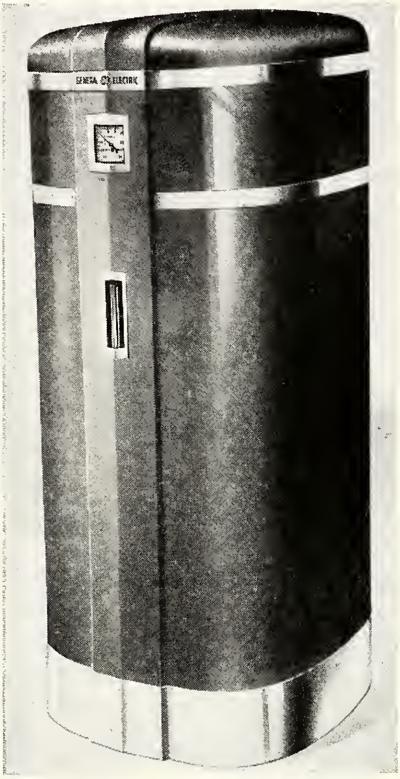


Fig. 4.14. External view of General Electric boiler. (*Courtesy of General Electric Company.*)

The General Electric Oil Furnace is the only example of its unique design now on the market. However, the wide popularity and good record of these boilers in service entitles the design to our particular attention. Two illustrations are provided; Fig. 4.14 shows the boiler as the owner sees it, and Fig. 4.15 gives the internal arrangement.

There is an interesting division of the combustion air. A portion of the air leaving the built-in overhead fan is used to atomize the oil in the burner head; the remainder of the air travels through a duct to the bottom of the boiler, where it is introduced fountainlike into the bottom of the furnace. This rising air meets the descending flame and the gases of combustion, thus achieving high furnace turbulence and good combustion.

Domestic water heaters may be attached to this boiler. Although not submerged in the main body of the boiler water which is a common location in heating boilers, the domestic unit is nevertheless concealed within the over-all casing.

The Magazine Boiler. In Fig. 4.16, we observe a steel boiler which incorporates a coal bin, so to speak, inside the unit. This coal-storage space can be obtained in sufficient size to fire the boiler automatically by gravity for as long as 24 hr.

These boilers generally are built for the small- and medium-sized demands, up to about 4,600 sq ft of equivalent direct steam radiation.

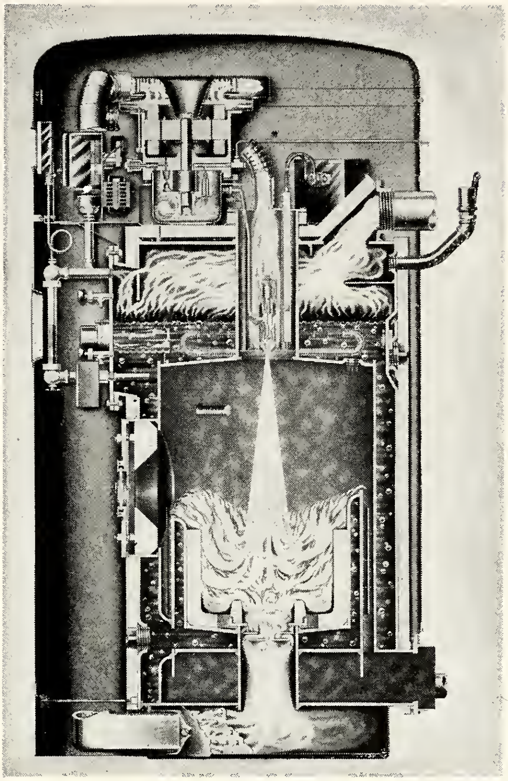


Fig. 4.15. Cutaway view of General Electric boiler, showing secondary air rising from bottom to meet flame and primary air. (*Courtesy of General Electric Company.*)

Gas-fired Boilers

General Features. Special attention should be given to gas-fired boilers because of (1) their ratings, based on standards set by the American Gas Association, (2) their need for the right fuel, and (3) their need for a special device, the draft hood.

Gas-fired boilers may be built of either cast iron or steel, similar in all their design features to boilers for any other fuel. We find the first variation in the furnace.

Since forced draft fans are not used, the furnace is largely open to the room, in residential and other heating installations. The gases of combustion pass through the boiler, but before reaching the stack, they must traverse a device known as the "draft hood." The draft hood has its underside open to the room, and its prime purpose is to permit the escape into the room of downdrafts from the chimney which might otherwise blow out the burners.

The room air induced into the chimney through the draft hood, when the combustion gases are passing normally to the stack, also inhibits the formation of moisture deposits by the gases. These moisture deposits if permitted are quite corrosive.

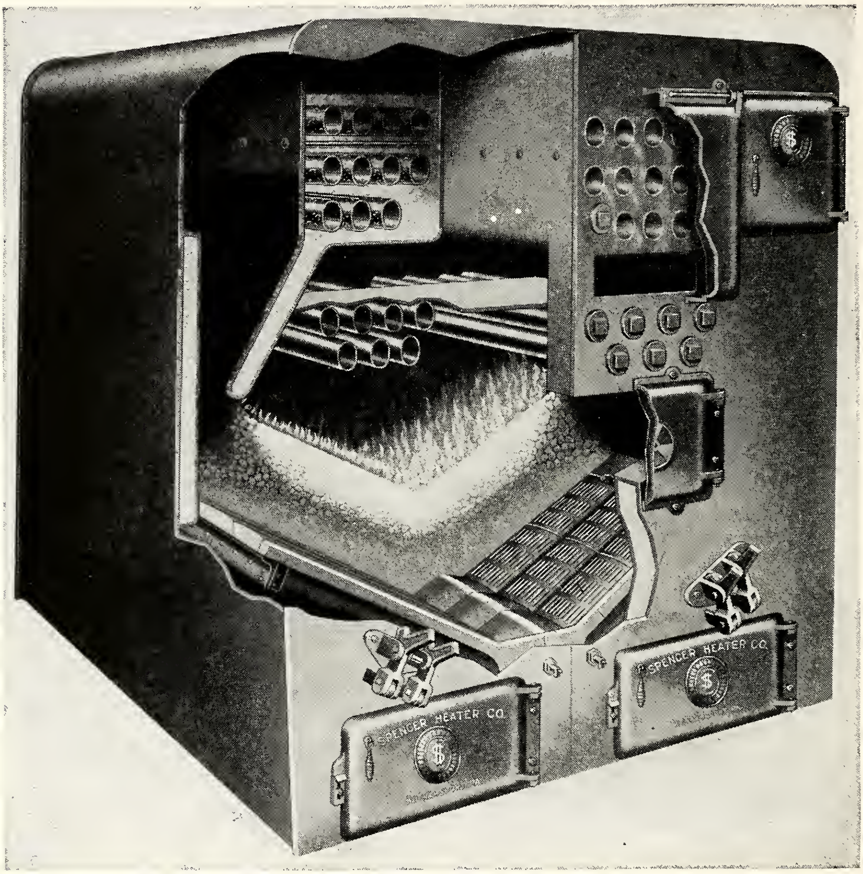


Fig. 4.16. Magazine boiler for coal firing. Magazine normally holds at least one day's supply of fuel. (*Courtesy of Lycoming-Spencer Division, Avco Manufacturing Corp.*)

Wrong fuel for a gas-fired boiler means, for example, the supplying of butane to burners designed for artificial gas, or vice versa. We cannot use any gas in any burner.

It also means using a liquefied petroleum gas, or rather trying to use a liquefied petroleum gas, that remains liquid and will not gasify in cold weather. The consequence is simply no gas and therefore no heat. We have the wrong gas.

In order to use more than one type of gas under our gas-fired boiler, we must provide a set, or several sets, of change-over burners, depending on the varieties of gas we contemplate using.

To see how a gas-fired heating boiler looks when delivered to the job, we examine Fig. 4.17. With the cover off, we can note the paths of travel for both the water and combustion gases in Fig. 4.18.

The unit pictured is designed for hot water systems only, with a capacity range between 57,600 and 400,000 Btu per hr output. The input varies between 72,000 and 500,000 Btu, respectively, thus indicating an efficiency of approximately 80 per cent.

To convert these Btu capacities into equivalent square feet of hot water radiation, we need only decide on a temperature for the water, and then refer to Table 4.2. For example, with 195 F water, the 400,000 Btu boiler should serve 2,000 sq ft of radiation.

Large power boilers fired with gas are not subject to the draft-hood requirement, since they are necessarily fitted with induced draft fans to produce a positive draft through the boiler.

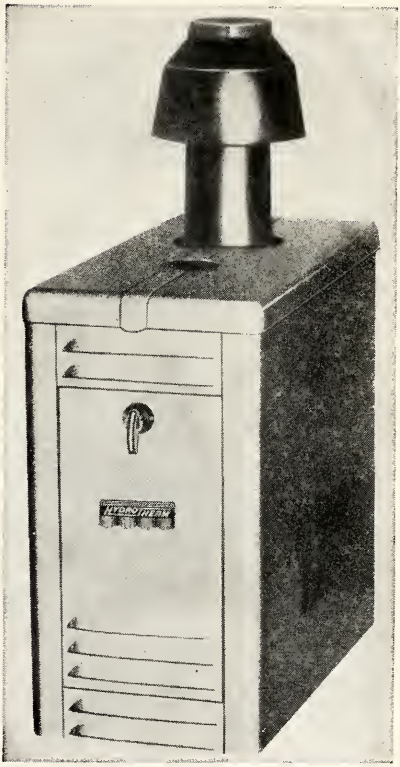


Fig. 4.17. External view of gas-fired hot water boiler. (*Courtesy of Hook & Ackerman, Inc.*)

Figure 4.19 shows a type of gas-fired boiler that may be used for either hot water or steam heating. This unit is available in very large sizes, up to 3,840,000 Btu per hr heat output. Converted into steam radiation, if we divide by 240 Btu per hr per sq ft of steam radiation, this unit should serve 16,000 sq ft.

An interesting feature of these large sizes is the employment of more than one draft hood. In each installation, the boiler builder will inform us of the number of draft hoods necessary. However, one correctly sized chimney flue is adequate for a multiplicity of hoods.

Figure 4.20 allows us to look into the interior of a large steam or water boiler. The iron sections are lined up together in an arrangement suggestive of the conventional cast-iron sectional boiler, but the details of sections designed specifically for gas differ from those for other fuels. The reason is that with gas fuel we have smaller volumes of combustion gases to handle and at the same time the gases are of lower temperatures.

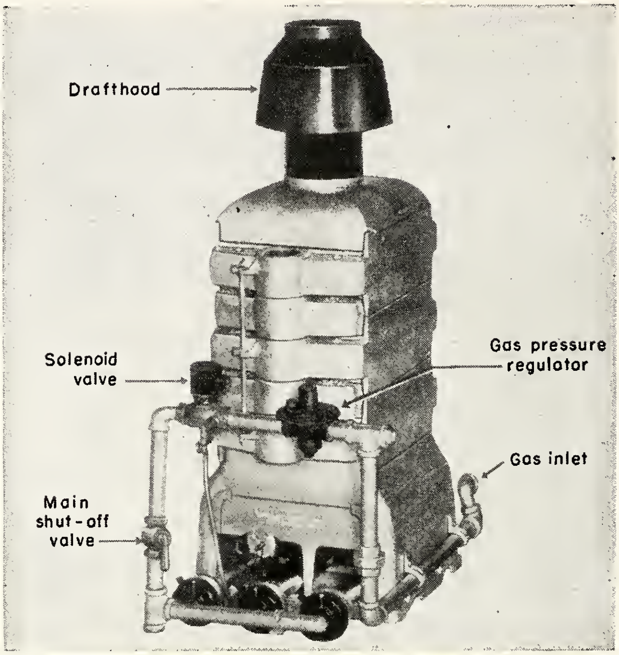


Fig. 4.18. Gas-fired hot water boiler with casing removed. Note horizontal arrangement of cast-iron sections. (*Courtesy of Hook & Ackerman, Inc.*)

The range in physical size of gas-fired boilers is given in Table 4.3.

Table 4.3. Gas-fired Boiler Dimensions

Available output, Btu/hr	Over-all dimension			Flue openings		Size of connection to stack, in.
	Length	Width	Height	No.	Size, in.	
57,600	2′2″	1′1″	2′2″	1	5	5
3,840,000	15′6″	3′7″	4′7″	4	16	30

These boilers, we observe, tend to be long and narrow and of modest height. These dimensions may be found to be ideal for some installations. Figure 4.21 shows a typical burner grid assembly.

The American Gas Association Ratings. The possible variations of the calorific values of the gas fuels we buy for our gas-fired boilers has led the American Gas Association into running tests to determine just and proper ratings. Obviously, if we say our boiler can serve 2,000 sq ft of radiation with a gas of 1100 Btu per cu ft supplied to the burners, a reduction in the calorific value of the fuel gas to, say, 550 Btu would leave us short of heat. In everyday practice this reduction can be brought about simply by changing over from a natural gas, or a liquefied petroleum gas, to the ordinary manufactured city gas.

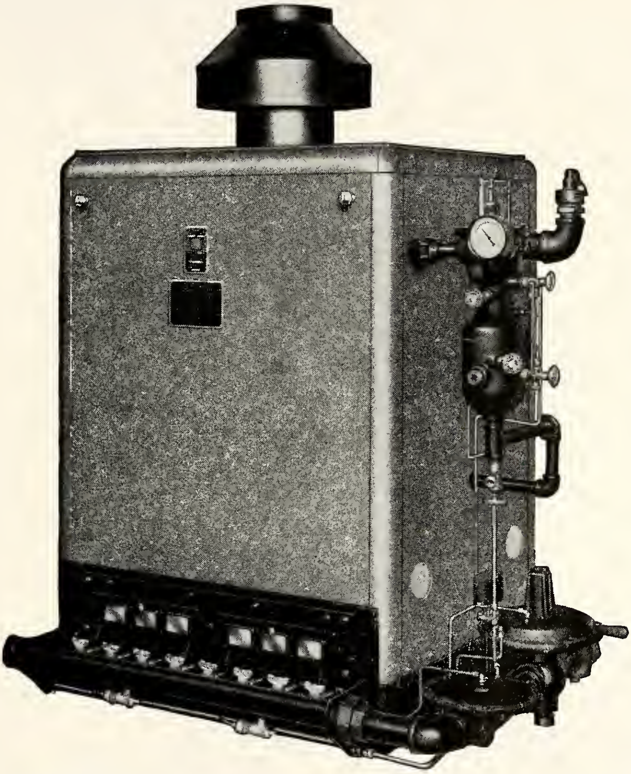


Fig. 4.19. Gas-fired steam boiler with gas main kept clear of casing. (*Courtesy of Pennsylvania Furnace and Iron Company.*)

In consequence of this situation, the American Gas Association stipulates that a boiler's capacity should be stated in Btu per hour output, based on the Btu per hour input.

In using the AGA standards with our calculations, we first determine the heating load in Btu per hour. This load should include the radiation, the allowance for piping loss, for pickup, for domestic hot water, and for any other requirement that makes us burn fuel.

With this total, we next refer to a manufacturer's catalogue and select

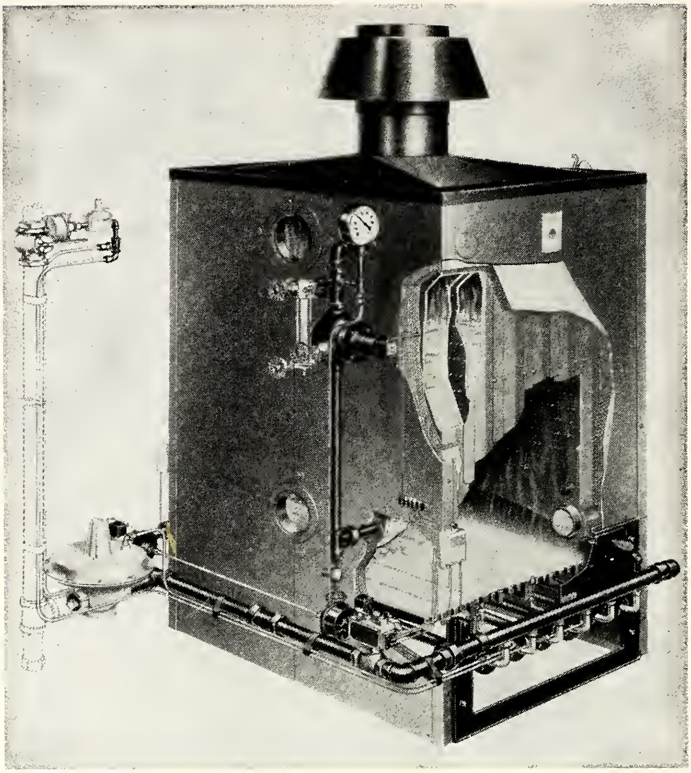


Fig. 4.20. Gas-fired steam or water boiler with vertical cast-iron sections. Note flat arrangement of burners and draft hood on top of boiler. (*Courtesy of American Radiator and Standard Sanitary Corp.*)

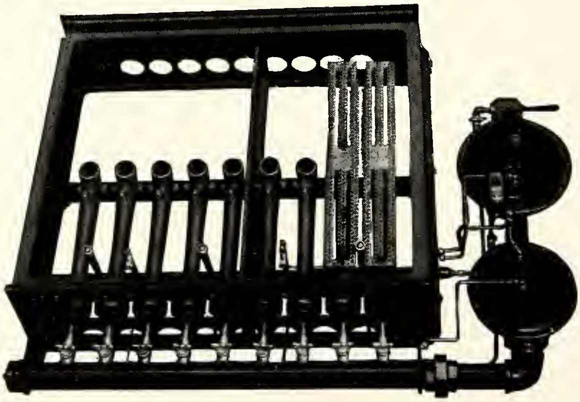


Fig. 4.21. Detailed view of burner grid. (*Courtesy of Pennsylvania Furnace and Iron Company.*)

the boiler which is listed as making available the number of Btu we need. We next consider the Btu input, and with this information are able to size our fuel piping, flues, and chimney.

The output in Btu can be easily converted if we wish into equivalent steam or hot water direct radiation.

Steam Generators

General Features. Technically, any vessel in which water is made to boil is a steam generator. Under this definition many of the cast-iron and steel low-pressure heating boilers would be steam generators.

The term has been given a different meaning, however, in late years, largely because of dissatisfaction with the practice of expressing boiler capacity as boiler horsepower.

A boiler horsepower, according to the definition formulated in 1915 by the Power Test Code of the American Society of Mechanical Engineers, represents the evaporation of 34.5 lb of water from 212 F into dry, saturated steam at the same temperature. This definition superseded an earlier concept which set the weight of water as 30 lb and the conditions of evaporation as 100 F for the water and 70 psig pressure for the steam into which it became converted.

Nowadays, the ASME definition is heard generally as "34½ from and at," an example of the American desire to shorten everything.

Expressed in Btu, a boiler horsepower means 34.5 × 970.3, or 33,475 Btu. The figure 970.3 in the equation is the latent heat of evaporation at 212 F under atmospheric pressure.

In 1915, the ASME definition seemed quite acceptable, but even at that time the modern steam generator as we know it today was entering the design stage. Then boilers were all of riveted construction, and working pressures seldom exceeded 275 psig, due to many as yet unsolved problems of boiler materials and their joining. For instance, the steel alloys that we use today for working temperatures of 900 F to 1000 F were still a dream; at these temperatures the steel glows redly in the dark, and 35 years ago the sight would have been considered a prelude to disaster.

As materials, designs, and construction improved, pressures and temperatures began to climb. Suddenly it was realized that boiler output might conceivably depend on the firing equipment and on the facility with which water could be made to circulate through the tubes. Boilers rated at any given horsepower were rerated to much higher values simply as the result of improved firing equipment, or improved firing conditions.

The custom of saying that any given boiler was operating at from 200 to 400 per cent of its rating merely indicated that the boiler was delivering from two to four times the amount of steam officially considered its normal capacity by the builder. All this led to much smaller boilers producing much more steam than they had ever produced before.

The situation now made the selection of the firing equipment more or less essential to the boiler builder. Obviously, a boiler of small size when sold as a, say, 1,000-horsepower (hp) unit, could only deliver 1,000 boiler horsepower (boiler hp) when properly integrated with the stoker, oil burners, or other firing equipment. The forced and induced draft fans likewise became critical factors in any over-all guarantee.

The next step became a very natural one. The boiler manufacturer began offering a complete steam producing plant, and expressed the capacity of his product, not in boiler horsepower and percents of rating, but in pounds of steam generated per hour at the pressures desired by the purchaser, and under the conditions of water and fuel provided by the purchaser.

In this complex and scientifically designed steam producer, the boiler obviously became merely a component. A more comprehensive term was needed, a term that did not depend on physical heating surface areas, or on an arbitrary ASME definition applicable to only part of the whole. Thus the tern "steam generator" came into use, and the "normal capacity" of the generator is exactly the capacity for which it is designed.

This is the steam generator we find in public utility power plants, and in industrial plants where high pressures, high temperatures, and large steam demands must be satisfied. The steam generators met in heating practice and in small industries are quite different in many points, and are generally termed "package generators."

Package Generators. Package generators, like power-plant equipment, are built and sold by the boiler manufacturer as a unit, or "package." The buyer merely connects up the water, electricity, oil or gas piping, steam piping, and stack. He lights the burner, and his package generator begins to generate steam.

The package generators resemble Scotch boilers in having a cylindrical furnace, located at or below the center of the boiler, and gas tubes submerged in the water. Three passes are common in the gas travel, which means that the gases go back and forth three times through the tubes before reaching the stack.

Figure 4.22 illustrates clearly the arrangement and workings of the interior of the boiler.

Since it is very difficult to provide strength in a large drum without going to metal of excessive weight, the package generators are at present limited in their size. The term "boiler horsepower" is still commonly used to describe the capacity of these generators, and in this case it is a reasonably exact term, for this type of unit cannot be overloaded due to limitations of furnace volume and circulation crises in the drum. In short, we can burn just so much fuel in any given unit and no more; the furnace won't allow more, and the natural circulation slows down in the drum owing to the equalization of water temperatures. Formation of steam then occurs in big bubbles that may belch water up and into the steam piping.

For heating installations, the package generators are designed for pressures of 15 psig. If we need higher pressures for process work, we can buy a generator for 100 lb or more, and step down the heating steam through a pressure-reducing valve.

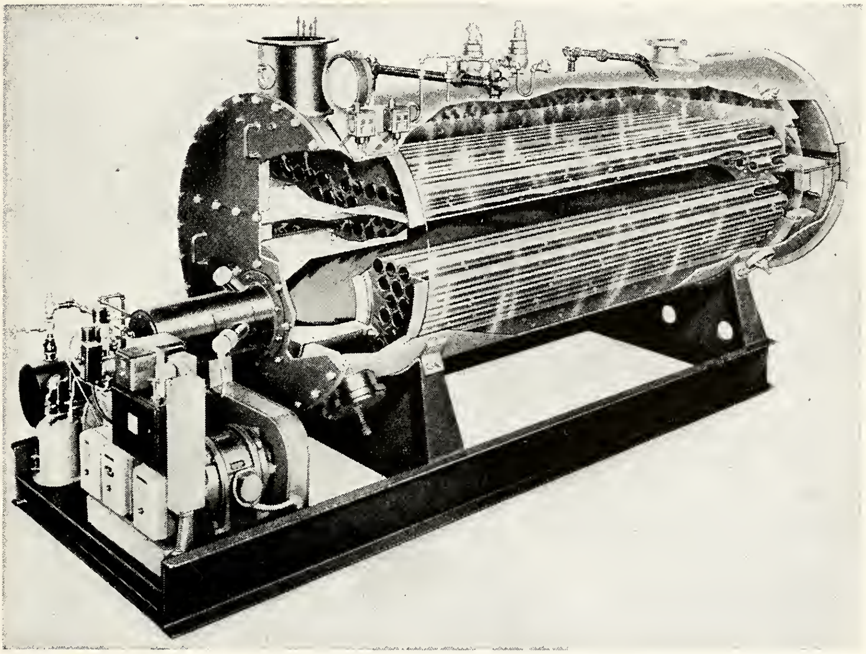


Fig. 4.22. Package steam generator, using fire tubes for flue gases. (*Courtesy of Cleaver-Brooks Company.*)

The largest package generator now available is 500 hp, and not all of the builders are prepared to furnish one this large. For all standard installations, 300 hp is the common top, presently available from all the well-known manufacturers.

Figure 4.23 shows a package generator of another type, a water-tube boiler being the heart of the design.

At present, the package generators are fired with oil or gas. The small size and cylindrical shape of the furnace tends to make stoker firing impractical, and of course removal of the ashes would be a problem. However, the use of pulverized coal is not impossible, and we may see such firing before too long.

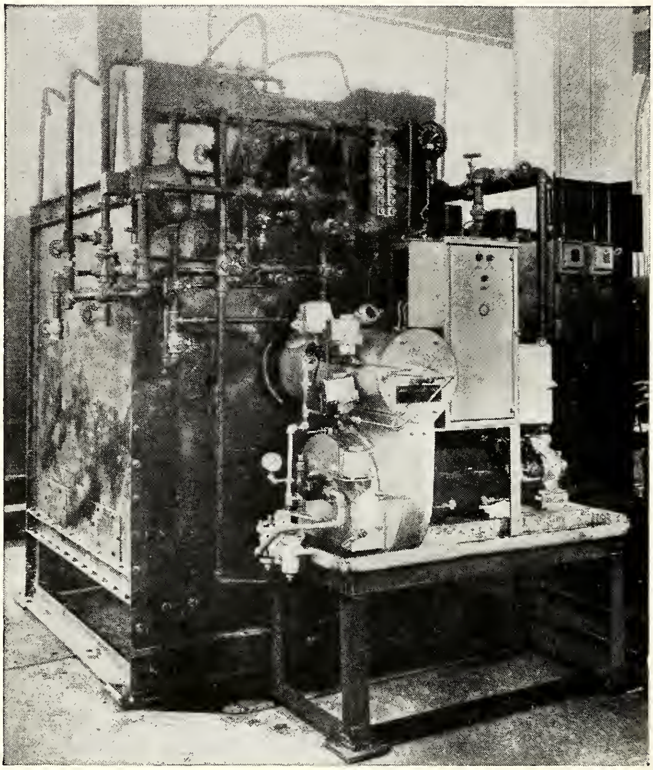


Fig. 4.23. Package steam generator, water-tube type. *(Courtesy of Heilman Boiler Works, Inc.)*

Special Forms

Waste-heat Boilers. Most of us never will be concerned with reclaiming waste heat as a source of steam or hot water for our radiators. However, we should know that such a practice exists, and should have some idea of what a waste-heat boiler is like.

Actually, a waste-heat boiler is more of a heat exchanger than a boiler, because it has no furnace of its own. The hot gases which pass through it originate elsewhere, in a furnace that is being operated for some reason other than the making of steam.

For example, in making artificial gas by the carbureted water gas process, forced draft is applied intermittently to a bed of burning coke which

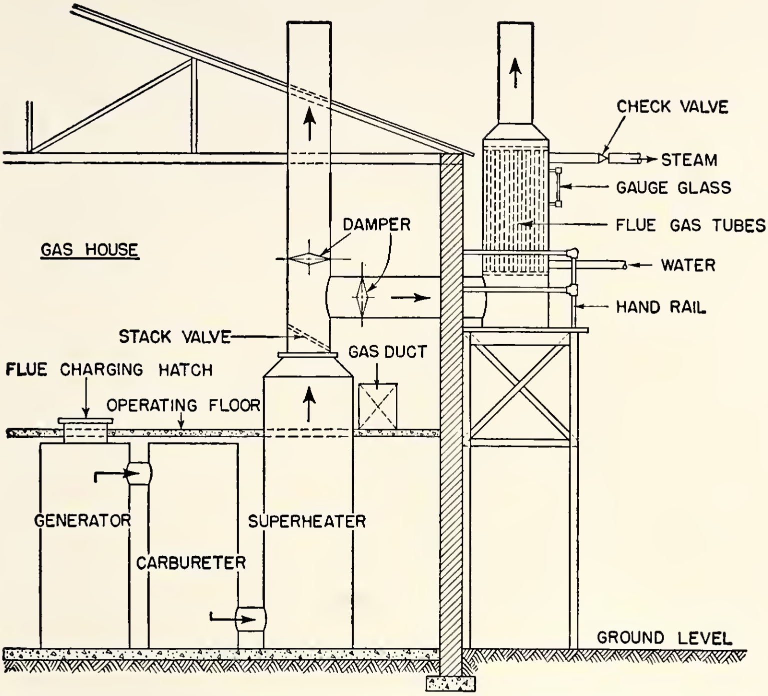


Fig. 4.24. Waste-heat boiler coupled to a water-gas generator.

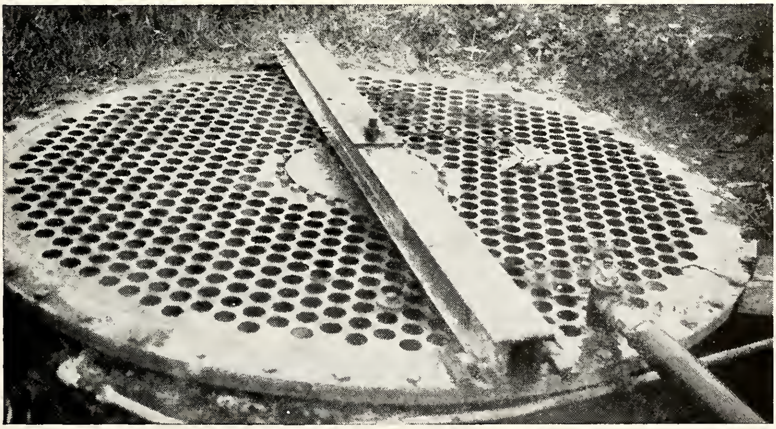


Fig. 4.24a. Waste-heat boiler in a volcano steam vent. Note water-pipe connections.

is inside the gas-making machine. During this period of the "blow" as it is called, the products of combustion pass through a stack valve to the atmosphere.

If we now divert these gases at the stack valve to a heat exchanger of some kind, through which they must pass before reaching the atmosphere, our heat exchanger becomes a waste-heat boiler. Water will boil into steam from the Btu in the hot gas, and will continue to boil so long as the gas is passing through. Of course when there is no hot gas, there is no steam, and this is one of the characteristics of the waste-heat boiler in many industries—its production of steam is irregular.

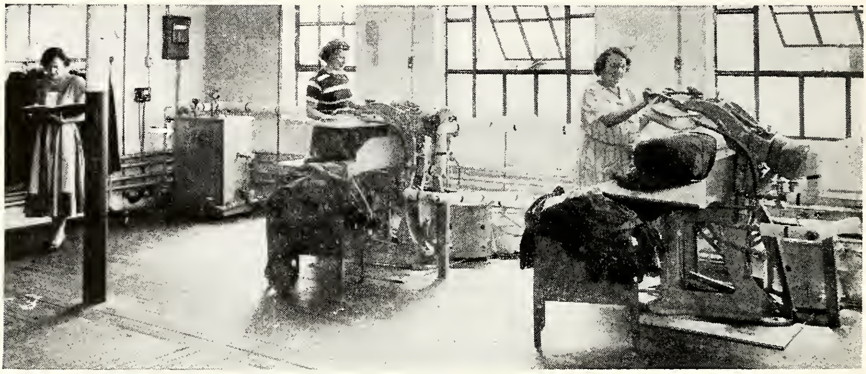


Fig. 4.25. Electric steam boiler. Note small size of this unit which is rated at 30 kw, or 100,000 Btu per hr. *(Courtesy of Livingstone Engineering Company.)*

Most waste-heat boilers are fire-tube units, since these are the simplest to install. Figure 4.24 illustrates a waste-heat boiler installation in a water gas plant. Figure 4.24*a* is of interest, because the boiler is set in the side of a live volcano, over a steam crack. This installation contributes hot water to the baths of the Volcano House, a hotel on the brink of the crater of Kilauea, in Hawaii.

Electric Boilers. Figure 4.25 illustrates one of the small electric boilers generally used for producing steam at high pressures, but which can be applied by the use of reducing valves to the heating loads of small to medium structures. The range of steam output varies between 45 and 300 lb per hr, at pressures of 15 to 500 psig, according to the design. The smallest unit carries an electric rating of 15 kw; the largest, 100 kw. Where electric rates are favorable, the use of such a boiler might be preferred by many persons.

Another type of electric heating contemplates the storage of electrically heated water in a large, insulated tank. The heating cycle is timed to take advantage of the lowest available electric rate, generally termed an "off-peak" rate. The heated water in the storage tank then is pumped to the individual radiators, convectors, or unit heaters, exactly as if it originated in a direct-fired boiler.

With this sort of scheme, the temperature of the circulating water steadily declines during the time the electrodes are nonoperating, and the radiation must be designed to meet the heating requirements of the building when using the water of lowest temperature. What this means to our square feet of standing heating surface can be calculated from Table 4.2. For instance, if we have a house with a calculated heat loss of, say, 100,000 Btu per hr and begin by circulating water of 215 F, the heat emission at this temperature is 240 Btu, and we will need 100,000 ÷ 240, or 417 sq ft. After some hours, the water temperature may fall to perhaps 160 F, and, since the heat emission at this temperature is only 130 Btu per sq ft, our house now needs approximately 769 sq ft.

Table 4.4. Boiler Fittings Supplied Normally as Standard Equipment

Kind of Boiler	*Fittings*
Hand fired, steam	Steam gauge, safety valve, water-gauge glass and trimmings, try cocks, automatic damper regulator, firing tools
Automatic firing, steam, oil or stoker.	Steam gauge, safety valve, water-gauge glass and trimmings, special platework for oil burner or stoker mounting, flue brush or scraper
Gas boilers, steam	Steam gauge, safety valve, water-gauge glass and trimmings, try cocks, flue brush, resetting low water cutoff, pilot, steam-pressure regulator, gas-pressure governor, draft hood
Hand fired, water	Combination altitude gauge and thermometer, automatic damper regulator, firing tools
Automatic firing, water, oil or stoker	Combination altitude gauge and thermometer, flue brush
Gas fired, water	Combination altitude gauge and thermometer, flue brush, pilot, water-temperature regulator, gas-pressure governor, draft hood
Package-steam generators: Heavy oil	Steam gauge, safety valve or valves, water-gauge glass and trimmings, automatic boiler feed with low water cutoff, vent pipe, 6 ft long, oil burner, fan or blower, combustion controls, control panel, flame-failure safety device, excess pressure cutoff, oil pump, oil preheater, oil-pressure relief valve, oil-pressure regulator, oil filter, electric-booster heater for fuel oil, oil gauges and thermometer, suction strainer, motors, motor starters, interunit wiring
Light oil	Generally as for heavy oil, except that preheating equipment is omitted
All boilers	Carefully check what firebrick is supplied, *if any.* Some manufacturers furnish all the refractory needed, some part of it. others, none

The economics of the storage system require careful study, for the advantages of the off-peak rate conceivably may be completely destroyed by the capital investment in tank and radiation.

Auxiliary Equipment

Boiler Fittings. When we buy a boiler, we may expect to receive certain fittings with the unit. If we want more than the manufacturer's standard practice includes, then we must buy these extra. The fittings vary with the heating medium, steam boilers being equipped with more items than are the hot water system boilers. The items also vary with the manufacturer, some units requiring special equipment as the result of their design. Typical fittings are given in Table 4.4 page 91.

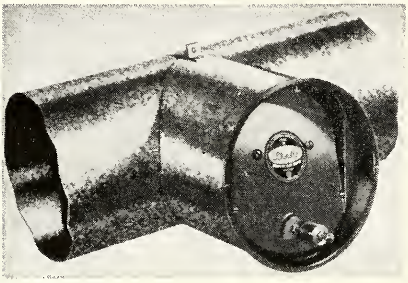


Fig. 4.26. Barometric damper as located in a boiler breeching. (*Courtesy of H. D. Conkey & Co.*)

Barometric Dampers. The barometer, an instrument for measuring the weight or pressure of the atmosphere, was invented in Florence, Italy, about 1643.

The barometric damper for controlling natural draft to boilers appeared in the mechanical market place about three hundred years later.

This damper is fitted into the smoke pipe between the boiler and the chimney. Its main feature is a side opening in the smoke pipe, with a balanced disk or gate that closes or uncloses this opening as a direct result of variations in air pressure between the boiler room and the chimney.

Figure 4.26 illustrates a barometric damper. The swinging gate is under the control of an adjustable counterweight, thereby permitting us to maintain the overfire draft at whatever percentage of the total draft we desire for best combustion. In practice, high chimney draft causes the gate to open, there is an inflow of room air to the smoke pipe, and thus the normal draft through the fire is not increased. Conversely, when the chimney draft is weak, the gate stays closed because the differential pressure is too small to move it, and the entire chimney draft is then placed at the service of the fire.

Boiler Insulation. All boilers should be thoroughly insulated. If insulation is not furnished by the boiler builder as a standard practice, then the designing engineer should see that his specifications call for it.

Figure 4.27 shows a sectional, cast-iron boiler covered with 85% magnesia, the surface finished off with a hard magnesia cement, troweled smooth. An 8-oz canvas jacket is specified by some engineers to go over the smooth magnesia surface, and omitted by others. The canvas preserves the integrity of the insulation against bumps, cuts, and similar destructive occurrences, but of course adds to the cost of the installation. Small residential jobs as a rule favor a metal jacket, factory applied.

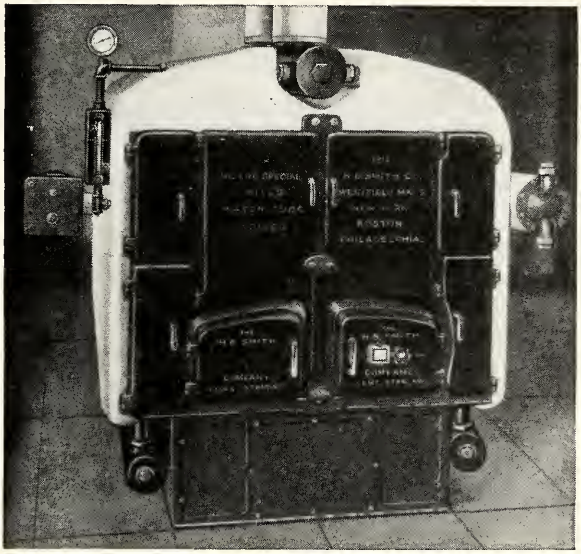


Fig. 4.27. Insulated finish of steam boiler in place of a jacket. Common in large units.

The thickness of the insulation depends on the temperature of the hot metal against which it is placed. Table 4.5 is provided as a guide.

Table 4.5. Thicknesses of Boiler Insulation

Temperature of Hot Surface, F	*Thickness of Insulation Exclusive of ¼-in. Finish Coat*
Up–250	1½ in.
251–350	2 in.
351–500	3 in.
501–650	1½ in. high-temp. blocks + 1½ in. insulation
651–800	2 in. high-temp. blocks + 2 in. insulation

While the table is set up for magnesia, it may be used for asbestos and combinations of asbestos and magnesia. In case of doubt, the manufacturer of the insulation should be consulted prior to writing the specification.

Altitude and Boiler Capacity

All the ratings discussed have been on the basis of sea-level installation. As the site of our boiler rises above the sea, some reduction in capacity takes place, but generally we need make no allowance for this fact under 2,000 ft. Above 2,000 ft the manufacturer should be consulted.

The exact mark down on any particular boiler depends on the design of the boiler. For example, one manufacturer sets a 20 per cent reduction on one of this model at 10,000 ft, but at the same elevation another model drops 45 per cent of its catalogue capacity.

What happens as we climb upward from sea level is this: We find the air density decreasing which means it is becoming thinner and thinner. With less air, there is less oxygen available for our lungs and less for combustion in a boiler. Since the amount of oxygen we need to support life or to burn a pound of fuel is reasonably constant, we simply must breathe faster or deeper to keep going ourselves, and must furnish more cubic feet of air per minute to the boiler furnace.

Table 4.6. Boiler Capacity at Various Altitudes, in Per Cent*

Altitude, ft	Barometric press., in. Hg	Air pres., psf	Air volume, cu ft/lb	Boiler capacity, %
Sea level	29.92	14.7	12.388†	100.00
1,000	28.86	14.2	12.834	96.40
2,000	27.80	13.67	13.268	92.90
3,000	26.76	13.16	13.701	89.40
4,000	25.76	12.67	14.111	86.1
5,000	24.89	12.20	14.469	83.2
6,000	23.86	11.73	14.866	80.00
7,000	22.97	11.30	15.237	77.00
8,000	22.11	10.87	15.621	73.90
9,000	21.29	10.46	15.956	71.20
10,000	20.58	10.07	16.253	68.80
15,000	16.88	8.32	17.802	56.30
20,000	13.75	6.74	19.078	46.00

* *Note:* This table should be used as an approximate guide only. Barometric pressures and local temperatures are subject to seasonal and sometimes hourly fluctuations, and consequently each installation should be rated only after consideration of local conditions at the point of installation.

† This is the volume of a standard United States atmosphere at 32 F and sea-level elevation.

However, there is a limit to the amount of air we can force through a boiler, and when we reach this limit, the boiler's ability to generate more steam or hot water stops right there. If this stopping point is 20 per cent under the catalogue capacity, then we must buy a larger boiler. Thus, to serve 1,000 sq ft of radiation at 10,000-ft elevation, we may need to select a boiler rated at 1,250 sq ft capacity.

Chimney design also is affected by altitudes. For the measure of this effect, we should refer to Chap. 16.

TABLE 4.7. AIR VOLUME AT VARIOUS ALTITUDES*

Altitude, ft	Equivalent barometric press., in. Hg	Air press., psia	Relative volume, cu ft/lb
Sea level	29.92	14.7	1.000
1,000	28.8	14.2	1.035
2,000	27.7	13.6	1.081
3,000	26.7	13.1	1.120
4,000	25.7	12.6	1.166
5,000	24.7	12.1	1.215
6,000	23.8	11.7	1.257
8,000	22.1	10.8	1.360
10,000	20.4	10.0	1.470
12,000	18.9	9.4	1.565
15,000	17.0	8.3	1.770

* *Note:* Both the relative volume and the density of the air are assumed in this table to be at constant temperature. Consequently, in using the table for combustion and stack-draft calculations at various altitudes, we must correct the air for temperature differences also. Table 2.7 provides information on the density and volume of air as affected by temperature. The general gas equation, which, of course, may be exercised at any time, is

$$V = \frac{1{,}544 \times \text{absolute temperature}}{\text{absolute pressure}}$$

Absolute pressure in this case means absolute pressure per square foot. Thus we convert the atmospheric pressure in pounds per square inch by multiplying by 144.

Questions

1. Describe the salient points for rating boilers under each of the following authorities:
    a. The Institute of Boiler and Radiator Manufacturers.
    b. The Steel Boiler Institute.
    c. American Gas Association.
2. What are the allowable working pressures for cast-iron boilers?
3. What is a wet-base boiler?
4. Describe the fundamental difference between fire-tube and water-tube boilers.

5. Describe the arrangement of a Scotch boiler.
6. Describe a firebox boiler.
7. What is the purpose of a smokeless arch, and how does it function?
8. Describe the operation of a downdraft furnace.
9. Tabulate the especially desirable features observable in each form of boiler described in this chapter.
10. How do installation requirements differ for a gas-fired boiler compared with boilers using other fuels?
11. Where and why would you specify a draft hood?
12. Explain to a friend the current meaning of the term "steam generator."
13. Describe the characteristics of a package generator.
14. For your own home, what type of boiler would you prefer? Why?
15. Describe the effect of altitude on boiler ratings.

CHAPTER 5

DESIGNING A HOT WATER HEATING SYSTEM

Natural, or Gravity, Circulation Systems

The Cause of Natural Circulation. If we fill a water bucket designed to hold exactly 1 cu ft, using water of say 200 F temperature, and then weigh the bucket, we discover our cubic foot of water has a net weight of 60.13 lb. If we next allow the water to grow steadily cooler, we observe that it seems to shrink, so that when it reaches a temperature of 52 F we must add water to fill the bucket. By using the scales, we find our cubic foot of water weighs substantially more than did the hot water, having now a net weight of 62.42 lb.

This temperature of 52 F is an interesting one, for here the density of water stabilizes, no further shrinkage takes place, and at 32 F just before freezing it still weighs 62.42 lb to the cubic foot.

Natural circulation, or gravity circulation, to use the term we hear more often in heating problems, obtains its motive force from the difference in density that results from the differences in temperatures. Figure 5.1 illustrates what happens in a hot water heating circuit. The column of 150 F water in the return weighs 61.20 lb per cu ft, while the 180 F water leaving the boiler weighs only 60.57 lb per cu ft. Naturally the cooler water pushes the warmer water upward from the bottom of the boiler, being constrained to do this by its superior weight. All we need do is to supply heat to the boiler, and our water goes around and around, from boiler to radiator to boiler again, the loss of heat in the radiator supplying the difference in density.

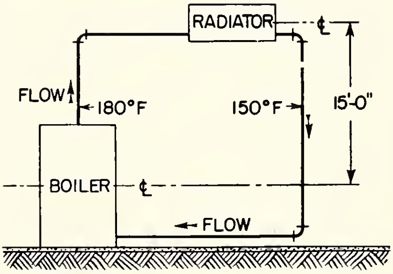


Fig. 5.1. Elementary circuit diagram, hot water system.

Calculating the Motive Force for Natural Circulation. Being generated by a temperature difference that averages a mere 30 to 40 deg in everyday heating practice, our motive force is of slight power, at best.

For instance, refer to Fig. 5.1, the difference in weight between the 180 F supply and 150 F in the return is only 0.63 lb per cu ft. Expressed in pressure per square inch, this means 0.63/144 sq in. = 0.0044 psi.

Such a figure is awkward to work with; therefore we use another measure, the *milinch.* The milinch is 1/1,000 of an inch, or 1/12,000 of a foot. Since a pressure of 1 psi is developed by a column of water 2.3 ft high (27,600 milinches), we can discover easily the pressure of 1 ft (12,000 milinches) by 1 lb/2.3 ft = 0.434 lb.

Next, to convert our 0.0044 lb pressure into milinches, we solve the simple equation

$$\frac{0.0044 \times 12{,}000}{0.434} = 122 \text{ milinches}$$

This equation is based on common logic, for if 0.434 psi is exerted by a column of water 12,000 milinches high (1 ft), then our 0.0044 lb must be exerted by a column shorter in proportion. We need only set down the proportion, 0.0044 over 0.434, multiply by 12,000 milinches, and there is the answer.

Thus we see the difference in temperature between 180 F and 150 F can exert a circulating force of 122 milinches.

Relative Heights and Circulation. Fortunately for hot water heating systems, our radiators are usually well above the boiler, sometimes two or three stories. As a result the motive force is considerably increased. For example, our 122 milinches which developed from a column of water 1 ft high, grows to 2,440 milinches if the column extends to a radiator on the second floor, say 20 ft up. At the third-floor level, the pressure differential becomes 30 × 122, or 3,660 milinches.

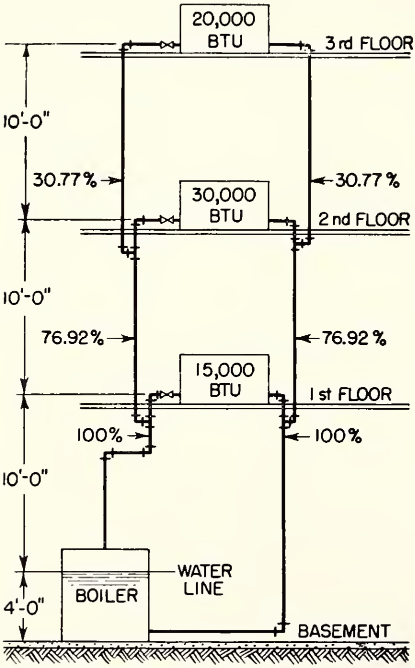


Fig. 5.2. Typical riser circuit.

At this point, we might stop to consider a natural phenomenon that affects our design of a gravity system. Figure 5.2 shows us three radiators, set one above the other on three different floors. On the first floor, we have a pressure differential based on 122 milinches multiplied by 10 ft, these 10 ft being assumed as the height of the radiator above the boiler. The total force available on the first floor is therefore 1,220 milinches.

Now we already have observed that the differential on the second floor is 2,440 milinches, and on the third floor, 3,660 milinches. The upper floors therefore, in consequence of having greater circulating forces to work with, will "hog" the heat unless we do something in our design to prevent it.

Several means are available for keeping all the heat from going to the top. The best is to design the piping and circuits so that greater resistances at the upper levels force the proper diversion of flow at the lower levels. A practical example of how this is accomplished appears in the following sections on sizing the piping.

Sizing the Gravity System Piping

General Considerations. Every foot of pipe, every fitting, every valve offers some more or less effective objection to the passage of water through it. The sum of these objections, expressed in our familiar milinches, must be kept below the available circulation-inspiring force. Thus, the maximum available force for the arrangement shown in Fig. 5.2 is 3,660 milinches, and all resistances must total less than this if we are to obtain circulation.

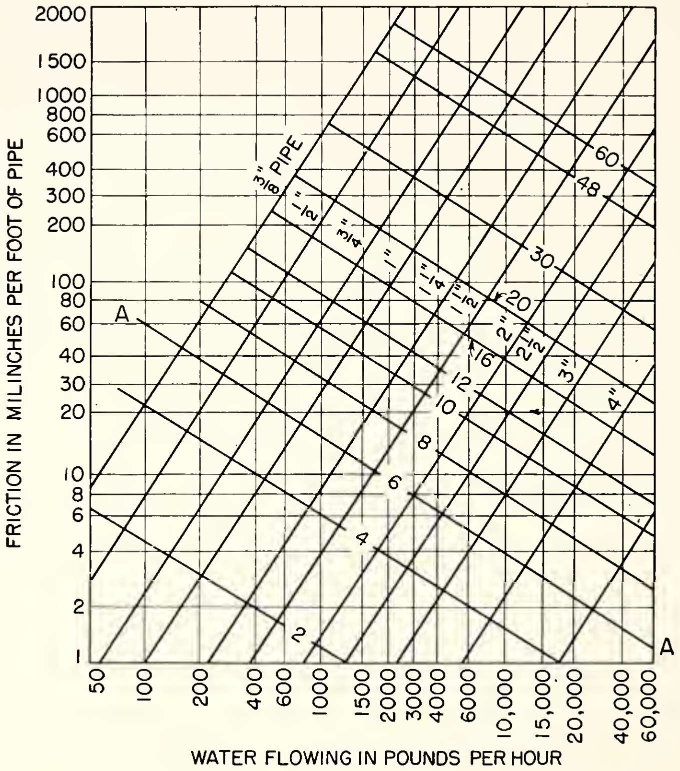
Referring to this figure, we have been wondering why our risers are not simply straight pipes, with radiator connections taken off through tees, and now we come to the explanation. We make the rising hot water pass through a couple of 90-deg turns to add resistance in the circuits to the upper radiators. In short, this is a means for making the circulation patronize the first-floor radiators in proper volume.

Our first step in sizing the piping is to find out how much water we need for each radiator. This is easy, for if 1 Btu will lift the temperature of 1 pound of water 1 degree Fahrenheit, then every pound which enters a radiator at 180 F and leaves at 150 F will drop 30 Btu during the passage. Consequently our three radiators will require water as follows:

Third floor	20,000 Btu ÷ 30 =	667 lb/hr
Second floor	30,000 Btu ÷ 30 =	1,000
First floor	15,000 Btu ÷ 30 =	500
Total		2,167 lb/hr

Water Velocities. Our next problem is to decide how fast the water shall move. Experience warns against designing for high velocities, because with gravity circulation high velocities are likely to be beyond us.

Suppose that we assume a velocity of 10 inches per second and see how this works out. Figure 5.3 which represents graphically the friction in iron pipes shows us that our total load of 65,000 Btu moving at 10 ips stirs up a resistance of 30 milinches per ft. Since we have 122 milinches available, we will run up the load line in Fig. 5.3 until we reach the 100-milinch level, as there is not any profit in selecting pipe larger than necessary. The velocity at this 100-milinch point is 16 fps and the pipe size is 1¼ in. We will accept these values until we check the pipe lengths and fittings and will tentatively figure on 1¼-in. piping.



NOTE: SLANT LINES A-A SHOW VELOCITY, INCHES PER SECOND

Fig. 5.3. Friction graph for pipe sizing. *(Abstracted by permission from the 1948 ASHVE Guide, Chap. 24.)*

The Effect of the Fittings. Table 5.1 gives the equivalent length of straight pipe for each of the commonly met fittings and valves. For various kinds of specialties, we must obtain the friction loss direct from the manufacturer. Thus, a cast-iron convector will produce a loss of one value, a copper convector of another shape a different value.

A look at Table 5.1 informs us immediately that we must calculate the

TABLE 5.1. FRICTION OF FITTINGS IN PIPE DIAMETERS

Fitting	*Equivalent Length of Straight Pipe*
90-deg elbow	25.0 × pipe diameter
45-deg elbow	17.5 × pipe diameter
90-deg long radius ell	12.5 × pipe diameter
Return bend	25.0 × pipe diameter
Open gate valve	12.5 × pipe diameter
Open globe valve	300.0 × pipe diameter
Angle radiator valve	50.0 × pipe diameter
Flow-control valve	500.0 × pipe diameter
Open stopcock	25.0 × pipe diameter
Tee:	
With 25% diversion to branch	400.0 × pipe diameter
With ⅓ diversion to branch	225.0 × pipe diameter
With 50% diversion to branch	100.0 × pipe diameter
With 100% diversion to branch	45.0 × pipe diameter

flow through each of the four tees in the system we are sizing, as the friction loss through a tee varies with the volume passing through the branch. The percentages through each tee may be determined by the following:

Third floor	20,000/65,000 = 30.77%
Second floor	30,000/65,000 = 46.15%
First floor	15,000/65,000 = 23.08%

While this method uses the loads in Btu per hour, the same percentages will result if we use pounds per hour or gallons per hour.

Looking at Fig. 5.2, we see that the first tee in the supply riser passes water for the two upper radiators, or a total of 76.92 per cent. The second-floor tee supplies the third floor only, and 30.77 per cent is the value at this point. Going across the radiators to the return piping, we find the same percentages to exist; thus for our friction calculations we have two tees rated at 76.92 per cent and two at 30.77 per cent. As the first two fall about halfway between the 50 and 100 per cent values in Table 5.1, we will assign them an appropriate friction value of 73 pipe diameters. The latter two are close enough to the 33 per cent line in the table to be given the loss of 225 diameters.

Since we have determined the value of the tees, we can now tabulate the whole line-up of fittings (see Table 5.2).

Final Sizing. Our next step is to multiply the total equivalent length by the friction loss per foot of the 1¼-in pipe as shown on the graph (Fig. 5.3), and the answer appears to be about 19,000 milinches. Therefore we cannot use 1¼-in. pipe. It creates too great a friction loss.

As we have an equivalent length of approximately 193 ft and a total head of 3,660 milinches, apparently a friction loss of 18 milinches per ft is permissible. On the graph (Fig. 5.3) an 18-milinch loss at 65,000 Btu per hr requires the use of a 2-in. pipe with a velocity of a little more than 8 inches per second. Perhaps our original assumption of a 10-in. velocity was not so far wrong at that.

Table 5.2

No.	Item	Equivalent pipe diameters	1¼-in. pipe, total equivalent length, ft (approx.)
11	90-deg elbows	275	29
1	Angle radiator valve	50	5
1	Radiator	75	8
1	Boiler	75	8
	Tees:		
2	76.92%	146	15
2	30.77%	450	49
	Total	...	114
	Total straight piping, from Fig. 5.2	...	79
	Total equivalent length	...	193

Now we must recheck our fitting evaluation for the larger size pipe, for as losses in fittings are expressed in pipe diameters, we can expect our equivalent over-all length to increase. The new picture is shown by the data in Table 5.3.

Table 5.3

Item	Equivalent pipe diameters	2-in. pipe, total equivalent length, ft
Ells	275	46
Radiator valve	50	8
Radiator	75	13
Boiler	75	13
Tees:		
76.92%	146	24
30.77%	450	75
Total	...	179
Total straight piping	...	79
Total equivalent length	...	258

At a rate of 18 milinches per ft, 258 ft total 4,644 milinches, which still exceeds our available head by 984 mil. The situation is now clear, and we shall be required to use 2½-in. pipe between the first-floor radiators and the boiler, on both the supply and return sides.

Right here we might note that the large piping required by natural circulation systems is one of the important reasons behind the development of the forced circulation layouts. Large piping is both expensive and space filling.

Just to complete the record, we observe that with 2½-in. piping our resistance per foot is a trifle more than 5 milinches and the velocity is 5 ips.

Sizing the risers to the radiators on the upper floors follows the same general plan, but the problem is much shortened and simplified by our knowledge of resistances and velocities. Starting at the first floor, we have a total load of 50,000 Btu per hr and a velocity of 5 ips. Referring to the graph, we observe that we must still use a 2½-in. pipe to the second-floor radiators, but the drop in load to the third floor is enough to permit us the use of a 1½-in. pipe for this last leg.

Valve-area Sizing Method Compared. There are other ways of sizing gravity system piping, generally much simpler, and generally less accurate. One of these, called the valve-area method, we will apply to Fig. 5.2 and compare the results with the outcome of our more lengthy procedure.

The principle of this method is to add together the areas of all the radiator valves on the main at any given point, and then make the main area at this point at least equal to their sum.

Our first step is to convert the output of each radiator into square feet of heating surface, then take the standard tapping for radiators of the sizes so determined, and add together their areas. Thus, with our design for 180 F water entering a radiator and 150 F leaving, we have an average 165 F in the radiator. According to Table 4.1 the heat emission per square foot of radiator surface will be about 140 Btu per hr.

Our 20,000 Btu load on the third floor, therefore, will need 143 sq ft of radiator surface; 214 must be provided for the 30,000 Btu load of the second floor, and the first floor has the modest requirement of 107 sq ft. From any catalogue of radiators we can find physical data, including tapping sizes approximately as shown in Table 5.4.

TABLE 5.4. SIZE AND AREA OF INLET

Inlet Size	*Area of Inlet, Sq In.*
143 sq ft, 1¼ in.	1.50
214 sq ft, 1½ in.	2.04
107 sq ft, 1 in.	0.86
Total area	4.40

By referring this area to Table 5.5 which shows us the dimensions of standard wrought-iron and steel pipe, we discover that the valve-area method also calls for a 2½-in. pipe to carry the whole load. For the first leg of the riser above the first floor, the radiator inlet areas total 3.54 sq in.; therefore we must again use a 2½-in. pipe. As for the third floor, a 1¼-in. pipe seems satisfactory and reasonable, but our more elaborate approach to the sizing problem indicates that a 1½-in. pipe is needed to keep velocities and friction within our available head.

Some designers feel that use of the valve-area system of sizing produces piping somewhat larger than is necessary. No doubt with some installations this belief may be substantiated; on the other hand, every system presents problems and characteristics that are peculiarly its own, and experience will oftimes guide us in our decisions. For example, a generally sound rule of thumb allows us to add 50 per cent to our straight pipe lengths for fittings and valves, but in the design just studied, fittings and valves actually added 153 ft of equivalent length to the 79 ft of straight pipe. On the basis of the 50 per cent rule, our conclusions would have been in serious error. Almost certainly we should have chosen piping too small for the job.

TABLE 5.5. DIMENSIONS OF STANDARD WEIGHT IRON PIPE

Pipe size, in.	Inside diameter, in.	Inside area, sq in.
½	0.622	0.304
¾	0.824	0.533
1	1.049	0.864
1¼	1.380	1.495
1½	1.610	2.036
2	2.067	3.355
2½	2.469	4.788
3	3.068	7.393
4	4.026	12.730
6	6.065	28.891
8	7.981	50.027
10	10.02	78.855
12	12.00	113.097

Sizing Horizontal Piping. We must exercise special judgment in sizing horizontal piping. Here we have no static head to aid the flow of water, and we must depend entirely on the pressure differential developed by temperature to move our water through the pipe. If the horizontal run is long, say over 10 ft, choosing a pipe one size larger than theory calls for will reduce friction and perhaps prove essential for satisfactory operation.

If there is one rule above all others that we should observe in designing a gravity system, it is this: Design with the utmost care, and then evaluate the design with the most exacting judgment.

Forms of Gravity Systems

All hot water systems are subject to classification according to (1) the sort of circulation employed, whether gravity or forced; (2) whether one-pipe or two-pipe; (3) whether direct return or reversed return.

One-pipe Systems. Since we are now examining the gravity systems, suppose that we consider the one-pipe layout, shown in Fig. 5.4. Here we see a system incorporated in one circuit, with the hot supply water being constantly cooled by admixture with the returns. The farther we travel from the boiler, the cooler becomes the water delivered to the radiators; consequently somewhere along the line we must provide more radiator surface to compensate for the cooler water.

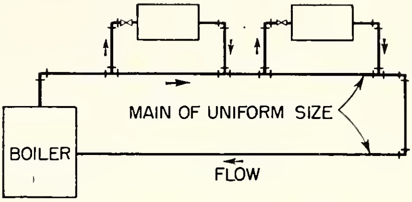


Fig. 5.4. Diagram of a one-pipe system.

This system is a simple one, but not wholeheartedly recommended except for the smaller applications.

We size the mains for a one-pipe system exactly as we did for the elementary system previously examined and sized. The radiator connections, however, introduce a complication in the form of progressively changing water temperatures and reducing differentials.

For example, we assume that a system is composed of five radiators with a total load of 50,000 Btu. We shall assume that the water leaving the boiler is at 180 F and that the return temperature is 150 F.

Now the first radiator will have 180 F water at the inlet, and since the outlet will be 150 F, the motive force in the radiator circuit will be realized from this 30-deg differential, or 122 milinches. We therefore size the piping to the first radiator on this basis.

The second radiator, however, is going to receive cooler water, as the result of dilution in the main from the 150 F discharge from the first radiator. This reduced temperature is calculated on the premise that each radiator in the circuit will contribute its share of the total 30 F drop in proportion to its own load. Thus, if we assume that radiator 1 emits 15,000 Btu per hr, the drop in temperature for radiator 2 will be $(15{,}000/50{,}000) \times 30 = 9$ deg. The second radiator therefore can expect to receive water at $180 - 9 = 171$ F.

The motive force to this second radiator is calculated on a difference of 171 F and 150 F, or 21 deg. Since water at 150 F weighs 61.2 lb per cu ft, and at 171 F, 60.78 lb approximately, our circulating head drops to 83 milinches. We must size the piping to the second radiator accordingly.

By the time the water reaches the third radiator, a further reduction of temperature has occurred. This time, on the assumption that the load in radiator 2 is 5000 Btu per hr, it will be

$$\frac{15{,}000 + 5{,}000}{50{,}000} \times 30 = 12 \text{ deg}$$

Radiator 3 therefore will receive water at 180 − 12 = 168 F. With a weight differential between 168 F and 150 F of only 0.37 lb, the circulating head for this circuit goes down to about 72 milinches.

Each of the radiators subsequently on the circuit must be considered similarly, and obviously the tendency of the water to circulate through the radiator weakens as the supply cools.

The Two-pipe Direct Return System. Figure 5.5 shows us an arrangement of piping that avoids mixing the returns with the supply water.

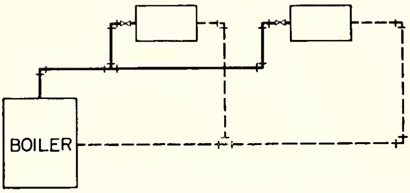


Fig. 5.5. Diagram of two-pipe, direct return system.

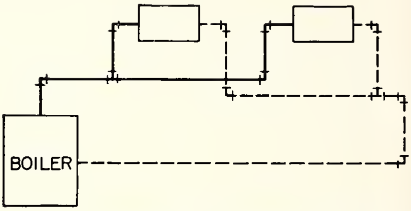


Fig. 5.6. Diagram of two-pipe, reversed return system.

However, this scheme takes little note of the inequalities of length which characterize the individual radiator circuits. In consequence, there is a tendency to short circuit, and radiators nearest the boiler when in the same plane may overheat while more distant ones underheat.

We calculate pipe sizes in accordance with the basic plan for this sort of system, which means we make no allowances for either the short circuiting or the probable inequalities of temperature. We just close our minds to facts, so to speak, and hope that nature somehow will correct the balance.

The direct return system does not enjoy very wide approval and should not be adopted if any other system can be made to apply.

The Two-pipe Reversed Return System. Here is undoubtedly the best design for a hot water installation. Figure 5.6 illustrates how a reversed return system is laid out. Basically, each radiator is part of a circuit which is approximately equal in length to all the other circuits. Thus, if the supply piping is of short run, the return is correspondingly long.

The reversed return is generally preferred by engineers for the medium and large installations, since balancing these larger jobs for the correct flow in all their parts becomes easier with this system.

However, the reversed return tends to add to the cost, and this additional expense is not always justified. Economics are important in heating just as they are important everywhere, and the designer must evaluate every job before he makes a final decision on the type of layout that will go in.

Occasionally it is possible to install a reversed return system without any additional cost, or perhaps a very slight additional cost.

Forced Circulation Systems

Benefits of Forced Circulation. By using a pump to circulate our hot water in place of the thermal head, we obtain several definite benefits.

First, we can reduce substantially the size of the piping, thereby saving money. Second, if we design properly, every radiator will get heat. Third, the use of a pump permits us to use hot water for panel and baseboard types of heating. Fourth, by using water almost at steam temperature, we can whittle down radiator sizes.

All these glowing benefits make us wonder just why anybody would choose gravity systems of hot water heating. Nothing, however, is perfect. A forced circulation system needs a pump which costs good dollars and is subject to the usual measure of casualties that befall operating machinery. If we must divide the system into zones or circuits, a common necessity for medium-to-large jobs, we may need a pump for every zone.

Pump failure can have serious results. As circulation stops, chilling starts, and if this unhappy condition occurs during the night or within a week end, frozen and ruptured pipes may produce for us a cold greeting at the first subsequent meeting.

The electricity that pumps use must be paid for, and while this is a small item, small items are of considerable importance to many persons.

All in all, we can think up numerous dangers and gloomy shortcomings of the forced circulation system, but the fact remains that a properly designed system provides excellent and dependable heating, and its popularity is increasing all the time.

First Considerations for Forced Circulation. In Figs. 3.3 and 3.4, we studied a steam heating layout for a two-story house. Now we shall design a forced circulation hot water heating system for this same house, and thereby enjoy the profitable thoughts that come from making comparisons.

Figures 5.7 and 5.8 therefore recall to us the room arrangement, dimensions, elevation, and other necessary data on which to base a heating design. Also, we must make some assumptions. First, there is the assumption of 20 deg for the temperature drop, and second, we shall set this drop between 205 F inlet water and 185 F outlet water. The average radiator temperature of 195 F will then give us a heat emission of 200 Btu/ft^2/hr, according to Table 4.1. The new radiator schedule, compared with the original steam schedule, now measures up as shown in Table 5.6.

The increase in heating surface for the hot water radiators is required of

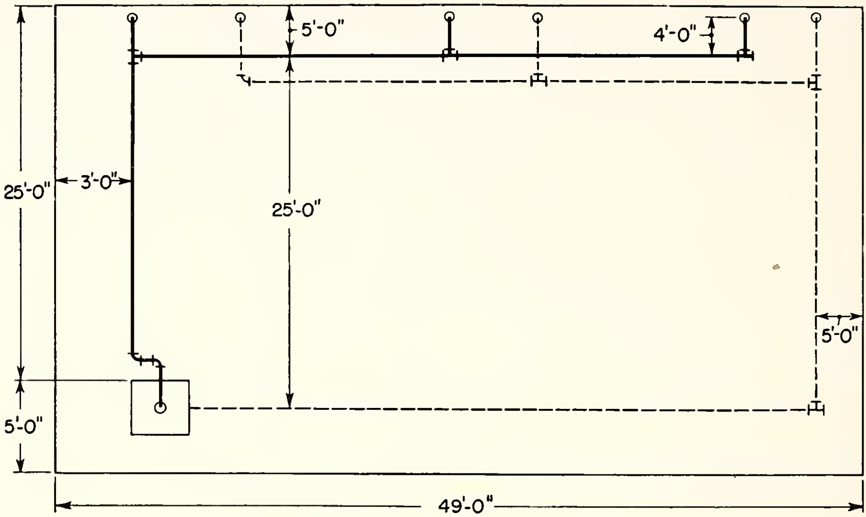


Fig. 5.7. Plan of two-story house.

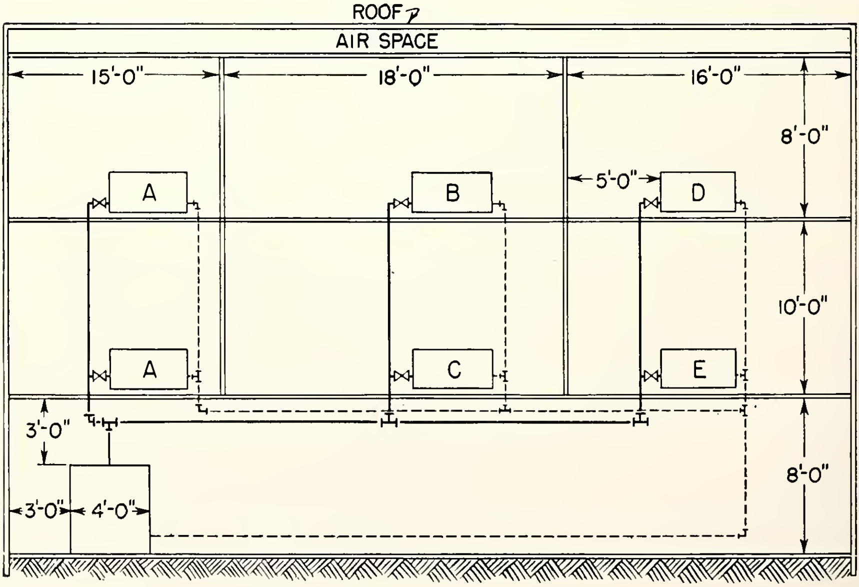


Fig. 5.8. Elevation of two-story house.

course by their lower temperatures. Emission from a steam radiator is calculated for most practical purposes at 240 Btu/hr/ft^2 of surface, and our 200 Btu emission therefore imposes on us the need to provide surface in the ratio of 1.2 to 1.0.

TABLE 5.6. NEW RADIATOR SCHEDULE

Symbol	No.	Heating surface, sq ft		Total sq ft of water
		Steam	Water	
A	2	100	120	240
B	1	60	72	72
C	1	45	54	54
D	1	120	144	144
E	1	30	36	36
Total..........		455	...	546

Just in passing we might observe that with gravity circulation putting out 150 Btu per sq ft per hr, the ratio is 1.6 to 1.0.

Sizing a Two-pipe Reversed Return System. There are several ways of designing a forced circulation system, all of which seem to come to the same successful end, but several steps are common to all.

First, we must determine the equivalent length of the longest circuit. In Figs. 5.7 and 5.8, this circuit starts at the boiler, runs to the *D* radiator on the second floor, and returns to the boiler. The measured distance, according to the dimensions in the figures, is approximately 73 ft supply and 74 ft return, a total of 147 ft. For fittings we shall allow 50 per cent, rather than calculating each one, as with a pump behind our water exactness is less critical than with a gravity head. Our total equivalent length therefore is 221 ft.

The next step is to decide on the amount of water that must be circulated. As the total load is taken for our purposes at 109,200 Btu per hr (546 sq ft × 200), and the temperature drop is 20 deg, 109,200 ÷ 20 = 5,460 lb per hr. Converting to gallons per minute is accomplished by another division using 480 as the divisor. This 480 is made up of 8 lb to the gallon for hot water ($8\frac{1}{3}$ lb is the familiar figure on cold water pumping problems) and 60 min to the hour. The answer is approximately 11.4 gpm.

Now that we know the procedure for calculating gallons per minute required by a system, we can simplify it for future use by setting the factors into a formula easy to remember:

$$\frac{\text{Total Btu per hour}}{\text{Btu emission per sq ft} \times 60 \text{ min} \times 8 \text{ lb}} = \text{gpm}$$

The next step in our design is to select a pump. Referring to the pump characteristic graph (Fig. 5.9), we observe that 11.4 gpm will be delivered by a 1-in. pump against a resistance of 4 ft (48,000 milinches). The pump that is the next size larger will give us 11.4 gpm at approximately 5.4 ft (64,800 milinches).

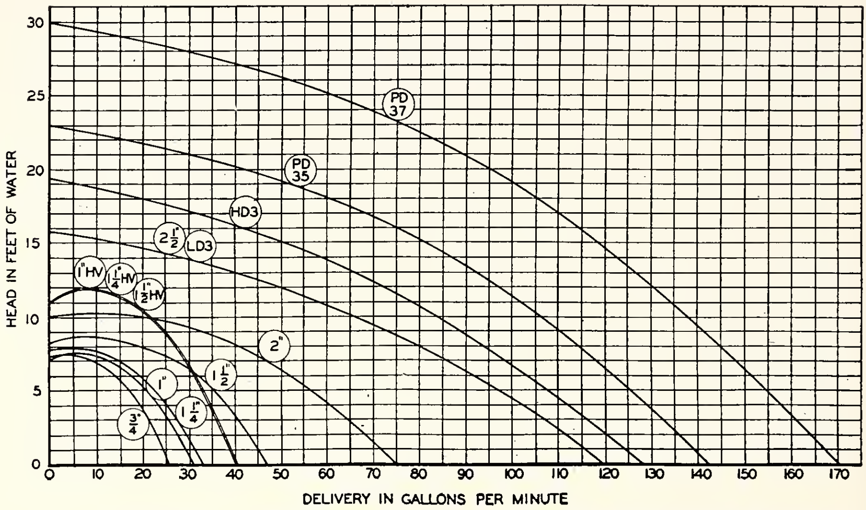


Fig. 5.9 Circulating pump characteristic curves. (Courtesy of *Bell & Gossett Company.*)

Which shall we use? If we divide 48,000 milinches by 221 ft, we have a design limit of 217 milinches per ft. According to the friction graph (Fig. 5.3), we can handle 109,200 Btu at this friction loss with a 1¼-in. pipe, but there is no factor of safety; we appear to be right at the critical point where maximum pump capacity exactly balances the maximum resistance.

Here we can do one of two things: (1) go to 1½-in. pipe, or (2) keep our 1¼-in. pipe but use the larger 1¼-in. pump. The larger pump with its 64,800-milinch head allows us a resistance of 64,800/221, or 293 milinches per ft.

As the difference in prices between a 1-in. pump and a 1¼-in. pump is not disconcerting, we will select the 1¼-in. pump for this job and keep the 1¼-in. piping from the boiler to the take-offs. The completely loaded return will also be 1¼ in. in size.

The time has arrived now to begin placing our findings on the piping diagram (Fig. 5.11). From the boiler to point *A* we shall use 1¼-in. pipe to carry 109,200 Btu. However, between *A* and *B* our load has dwindled to 61,200 Btu, and a look at the friction graph (Fig. 5.3) shows us we can use a 1-in. pipe for this leg of the circuit. From *B* to *C* our load is 36,000

Btu per hr, and, since this load is still too large for a smaller pipe, we must continue with the 1-in. pipe.

With the supply main sized safely, we turn our technical scrutiny on the risers. The first one, originating at point *A*, must satisfy a load of 48,000 Btu. If the same resistance of 293 milinches per ft is used, this load needs a 1-in. pipe to the first-floor take-off. On the second floor, the load is only 24,000 Btu, and the graph indicates that a ¾-in. pipe will do the job nicely.

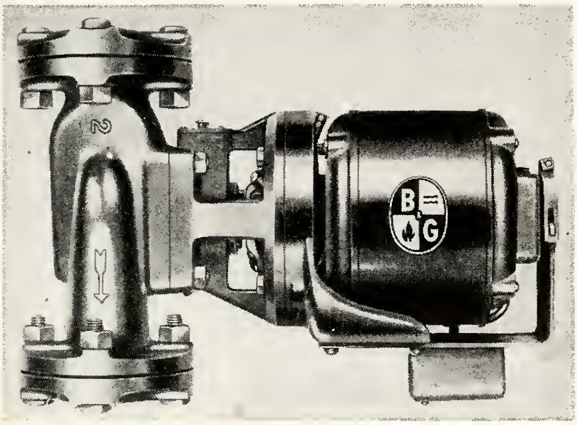


Fig. 5.10. Circulating pump. (Courtesy of Bell & Gossett Company.)

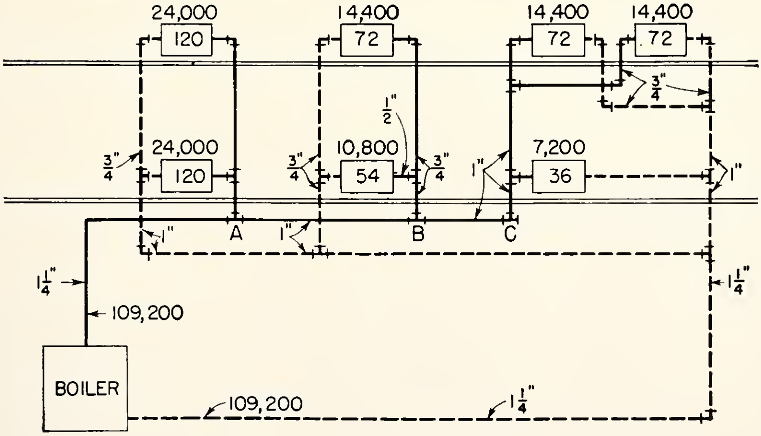


Fig. 5.11. Piping diagram for a two-pipe system, showing loads.

The same procedure is followed for the risers originating at points *B* and *C*, and the sizes taken from the graph are shown on the piping diagram.

The return is handled the same way. It starts with ¾-in. pipe at the outlet of the second-floor radiator *A*, increases to 1-in. at the junction with the first-floor radiator *A*, picks up the other returns in following its reversed route to the boiler, and ends up as 1¼-in. piping.

Table 5.10 (page 125) may be used to check our decisions on the graph selections, or we may skip the graph and take our sizes directly from the table. The advantage of the table is that it shows us *maximum* capacities for pipes of various sizes, at indicated friction loss per foot.

That big *D* radiator on the second floor will be far too bulky to please anybody with its 144 sq ft of heating surface; therefore our decision to make two radiators out of it will be applauded just as it was with steam. Each of the two substitutes will impose a load of 14,400 Btu per hr, so that to serve them our 1-in. riser must be divided into ¾-in. branches, as recorded on the piping diagram.

Making a piping diagram is essential for safe results. We can look at our sizes and loads, and occasionally pick up mistakes just because, on the diagram, the appearance of some figure or line strikes us as being out of balance with the rest of the system.

Radiator Connections. Radiator connections are sized from the same table, or graph. Thus radiator *C* on the first floor, with its load of 10,800 Btu, will require a ½-in. pipe.

Prudential Note. If convectors are to be used instead of the conventional cast-iron radiator, be sure to add in the resistance of the ultimate convector on the longest circuit, *as furnished by the manufacturer.* Some grievous experiences have resulted by the neglect of this often substantial resistance.

Designing a One-pipe Forced Circulation System. As in the case of the one-pipe gravity system, this layout delivers progressively cooler water to the radiators in any given circuit. One-pipe systems are easier to design to the extent that the mains are all of a uniform size, as all the water leaves from, and returns to, the same pipe.

Suppose that we take the same two-story house for which we just finished the design of a two-pipe reversed return system, and see what we can do with a one-pipe system.

Since we already know the load totals 109,200 Btu per hr and that we need a 1¼-in. pump to deliver 11.4 gpm with a system resistance of 293 milinches per ft, we are well along on the job. Of course, with a new problem, we face the necessity of making these computations to meet the new conditions, whether we contemplate a one-pipe or two-pipe installation.

On the piping diagram (Fig. 5.12) we shall set down the main size as 1¼-in. first of all. Now here comes a tricky phase of one-pipe design. Note well that the resistance in each riser circuit *must not exceed the resistance in the main between the points of take-off and return.* For example, if the distance between *a* and *b* on the piping diagram is 5 ft, then the total acceptable resistance in the riser circuit supplying two radiators is 5 × 293 or 1,465 milinches. The length of this riser circuit is approximately 26 ft plus 50 per cent for fittings, or 39 ft total. The per foot resistance then is 1,465 ÷ 39 or approximately 37 milinches.

The load on this circuit is 48,000 Btu, and for this load, both the table and the graph show that a 1½-in. pipe is the smallest to use at the points of take-off and return. For that part of the circuit supplying the second-floor radiator, where the load is only 24,000 Btu per hr, we can drop the size to 1¼-in.

The other two riser circuits are sized in the same manner. Between *c* and *d* we have only 3 ft, with a corresponding friction value of 879 milinches. The 6 ft between *e* and *f* provide us with 1,758 milinches. What we are able to do with these resistances is shown on the piping diagram (Fig. 5.12).

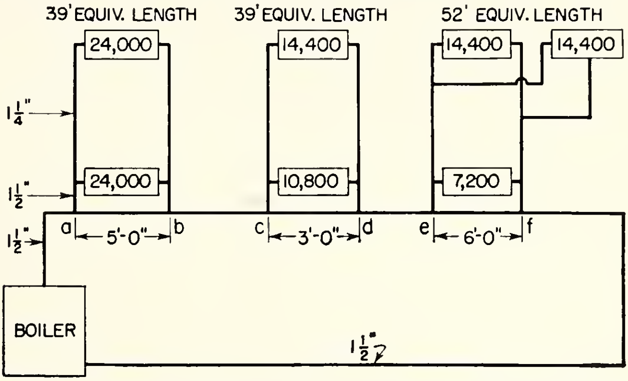


Fig. 5.12. Piping diagram for a one-pipe system, showing loads.

The use of 1½-in. riser take-offs from a 1¼-in. main is incongruous; therefore we will increase the main size to 1½-in. for this system. There is an advantage to the larger pipe main which compensates for the slightly slower circulation: It provides a larger reservoir of hot water and permits a somewhat better separation of the warmer and cooler water in the main.

Diversion Fittings. The name Monoflo is a proprietary one, and is used by the manufacturer to designate a type of fitting which diverts a computed volume of water from the main to the branches. Alternatively, we may obtain an adjustable tee for performing a similar service.

The importance of these fittings is primarily felt in system balancing, but they also in some cases permit the use of smaller risers. On occasion, we will be obliged to place radiators below the level of the main, perhaps set them on the basement floor, and to provide satisfactory circulation through these low radiators, the diversion fitting is absolutely essential.

Radiators below the Main. The difficulty in heating low radiators easily is understood. We are trying to make warm water, which is comparatively light in weight, go downward into a radiator already filled with heavier cold water, and there not only displace the colder water, but force it upward and into the circulation.

For example, suppose that we have a radiator 8 ft below the main. A column of, say, 200 F water, 8 ft high, exerts a pressure of 3.336 psi; a similar column of 180 F water produces 3.368 psi. The difference of 0.032 lb, or 883 milinches, is the static head which opposes the hot water as it endeavors to descend.

Actually the opposing force is much greater at times of starting up the system. After a week-end or even a one-night shutdown, the radiator temperature is the temperature of the room, perhaps 50 F. In this situation, the static difference between 200 F and 50 F water columns will be 0.132 lb, or 3,643 milinches, possibly enough to prevent the radiator from ever becoming heated.

The Monoflo fitting, or adjustable tee, is designed to divert sufficient pressure downward to establish the circulation. Conceivably, we can run into design conditions which present low radiators as the controlling system resistance, rather than the longest equivalent circuit. It is a situation we must watch for.

System Balancing. Trouble in a forced circulation hot water system is the outcome more often than otherwise of system unbalance. This is another way of saying that the resistances are not equalized, and short circuiting to a greater or less extent is occurring.

The designer can guard against this unpopular situation by the use of balancing cocks at strategic points, particularly the return junctions of extensive and irregularly loaded circuits. These cocks are merely square-headed devices which offer an adjustable resistance of their own to the flow of water through them. Their proper installation is in the return piping.

Another method for securing balance is to install orifices either in the pipe lines or in the branches to individual radiators. These orifices are commercially available, and buying is far simpler than attempting to make them. We tell the manufacturer the Btu we desire to pass and the temperature of the water together with the pipe size, and he will tell us how much friction the orifice develops at various velocities of flow.

Water Velocities with Forced Circulation. Experience indicates that water traveling over 4 ft per sec is not desirable. As for the friction loss, the most satisfactory range of design will be found between 250 and 600 milinches per ft.

Low velocities tend to nullify the fast circulation which is one of the advantages of pumping the water. On the other hand, excessive speeds of movement produce noise and high friction in the small pipes.

Locating the Pump. The pump may be connected into either the return or supply piping of the system. Some pumps are designed for mounting in vertical piping; some for horizontal installation. A few may be mounted either way.

Designers are more likely to set the pump on the return side for several reasons. One is that in this position it enjoys the full advantage of the static pressure of the cooler water. Also, because of handling cooler water, there is less likelihood of steam binding in case the resistance suddenly falls off and flashing occurs. However, in some of the modern cellarless homes, we may run into layouts with all the piping above the radiators, and in these instances the supply pipe offers the proper location for the pump.

Converting Gravity Systems to Forced Circulation

When we are asked to provide first aid in the form of a pump to an ailing gravity system, we go about the job in the following way:

First, we determine the load in Btu per hour. We should run off a check computation of the heat requirements for each room, and then if necessary increase the radiation. Fortunately, designers in the past seem to have regarded radiators rather pessimistically, and in consequence there is usually ample iron.

Sizing the pump can be dangerous, because we have no exact knowledge of the piping condition, and the selection of a pump that is too small will not earn us any commendations. After sizing the pump in the usual way, we should choose instead a pump at least one size larger, for experience indicates this is our only prudent course. If the existing system appears to be particularly doubtful, using a pump two sizes larger is not reckless, but safe.

Some of the old gravity systems will show up with long and involved runs. A good plan in these cases is to install a direct return to the boiler, but with a balance cock in route.

The technique of converting an old system includes a sizable dash of cut and try. If our first alterations are successful, we are merely fortunate, not expert.

Open and Closed Systems

General Characteristics. If we see an expansion tank overflowing from a neighbor's attic, the chances are that his house paint is being spoiled by an open system. Alternately, if the tank is located in the basement close to the boiler, with a relief valve that spills quietly down the drain when the tank pressure builds up to a certain value, then we have a closed system.

Figure 5.13 shows us typical open and closed tank arrangements, set side by side, so that the differences are clear without undue mental effort.

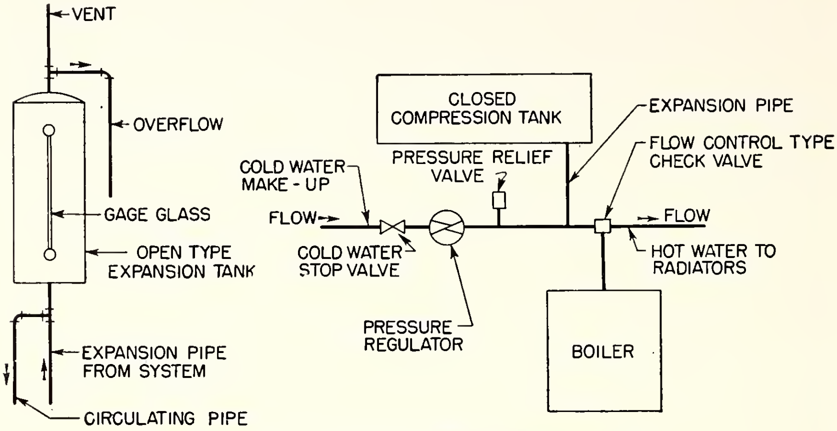


Fig. 5.13. Open and closed expansion tanks.

Sizing Expansion Tanks. The open system is open to the atmosphere at all times and must be arranged of course with the expansion tank higher than the highest radiator. To size the tank, we must calculate or measure the amount of water in the system when the system is *full and cold*. Between 40 F and 200 F, water expands to 1.04 times its original volume, and to provide for this expansion, our tank must accommodate theoretically 4 gal for every 100 gal in the system. Actually, we will size our tank for 6 gal per 100 instead of the theoretical 4, as once more the wisdom of experience is available to guide us.

Open systems may be used as either gravity or forced circulation installations, our major care being to see that we have the tank large enough, and high enough, to avoid squirting our hot water wastefully over the neighborhood.

Closed systems are pressure systems. As the water heats and expands, it compresses the air in the expansion tank, thereby making room for its own increased volume. Water will absorb air naturally and with particular alacrity under pressure. For this reason, draining of a closed system becomes imperative from time to time, in order to renew the charge of air in the tank. Alternatively, we might inject air into the expansion tank from a compressed air system, or even from a hand air pump if the latter can develop enough pressure. To keep the tank from rupturing under the strains of excess pressure, it must be fitted with a pressure relief valve.

Sizing Compression Tanks. In sizing compression tanks, we generally will figure on their holding 10 per cent of the water charge in the entire system. In large buildings, a tank of this size may become unwieldy and expensive; thus for these buildings we will use several tanks, locating them at strategic points. It is not unusual to see two or more compression tanks located side by side, above the boiler.

Table 5.7 makes available some figures that are helpful in sizing compression tanks.

TABLE 5.7. CAPACITY AND DIMENSIONS OF COMPRESSION TANKS*

Capacity of tank gal	For sq ft of radiation	Dimensions of tank, in.
18	500	13 × 39½
24	1,000	13 × 50½
30	2,000	13 × 61½
35	2,500	16 × 48
40	3,000	16 × 54
60	4,000	18 × 62
80	4,500	24 × 45

* *Note:* Tank sizes are based on forced circulation. For old systems use the next larger size of tank, or 10 per cent of the total water in the system.

Multiple Circuits and Zoning

Where Used. Buildings that are large, or rambling, nearly always require the application of several pumps to meet their varied heating demands. For instance, walls exposed to the south and east will require less heat to balance their losses in the morning hours than will walls having north and west exposure. Sunny areas are warmer than shaded, and these natural conditions affect the proper supplying of heat. Afternoon demands may be the reverse of morning demands.

Each area therefore must be considered in the light of its own peculiar conditions. Each circuit, or zone, must be designed to satisfy the load existing there, and must operate independently of all other zones. Excepting for the common boiler which supplies the hot water to all zones, each is in effect designed as a complete heating system with its own control.

Sometimes several zones with comparable heating requirements can be supplied by one pump; more often we will need a pump for each circuit. The selection of the pump is made in each case as though the circuit or zone which it serves is the whole system.

Tall Buildings. Buildings of many stories are served best as a rule if we sectionalize, or zone, them horizontally. For instance, we might consider the first four floors as one section, the next four as another section, and so on to the top.

If the building has considerable floor area or sharply contrasting exposures, then we may decide to zone each section further by providing two or more circuits.

Serving tall buildings from a hot water heating boiler in the basement is not practical, because the static pressure as the stories multiply soon exceeds the design limits of the boiler. We have already observed that a foot of water weighs 0.434 psi; therefore a 10-story building approximately 100 ft high would impose a pressure in the basement of 43.4 psi.

This would not be acceptable, since the standard limit for a cast-iron heating boiler is only 30 psi. In fact, as most city water systems carry a pressure of from 30 to 40 lb, a pressure-reducing valve must be placed in the fill line to our boiler to protect it against water-main pressure. This reducing valve is necessary for all hot water systems, both gravity and forced circulation.

The tall-building problem can be settled in any one of these ways:

1. Install a steam boiler and converters in the basement, the converters being designed to withstand pressures of 100 psig or more. Then the water is pumped from the converters to the various floors, while the low-pressure steam, condensed in the converters, returns to the boiler.
2. Install a steam boiler in the basement, and then locate the converters strategically on the upper floors. Thus a steam line would serve a converter on the fourth floor, another converter on the eighth floor, etc. Any other arrangement of the converters that seems suitable to the conditions of the job could be used, of course.
3. Install a low-pressure boiler on each of several levels. This multiplicity of boilers would not be cheap, however, as the chimney complications would grow, and there might be problems of fuel handling.

The converters are simply hot water generating tanks in which the heat is provided by a steam coil submerged in the tank.

Figure 5.14 illustrates schematically the idea of heating from converters in various locations.

Automatic Controls

Control of Circulation. Each hot water system requires a measure of automatic control with possibly one exception. This exception is the simple gravity system often used in small houses. Customarily with these simple installations, an increase in heat is obtained by going down into the basement and giving the fire more draft. Occasionally a clock or even a wire for hand pulling is connected to the damper, the first giving a modicum of automatic control while the latter saves climbing the basement stairs.

In forced circulation systems, hot water is delivered to the radiators only when the circulating pump is energized by a thermostat, a clock, or a throwing of the switch by hand. When the pump stops, circulation stops also, for a weighted valve on the boiler hot water outlet falls shut when the pump pressure is removed. The purpose of this valve is to prevent natural circulation from occurring and thereby nullifying our automatic control. In multiple circuit systems, we must install a weighted valve on each circuit.

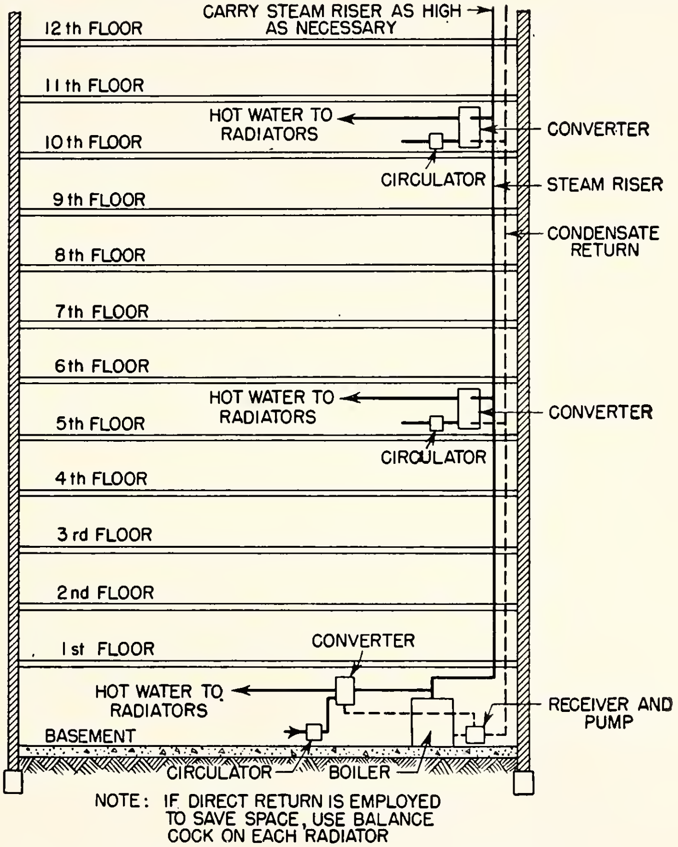


Fig. 5.14. Piping arrangement for a skyscraper.

These valves sometimes are called "flow control valves"; at other times and in other places designers call them back pressure valves. By either name they are simply check valves adapted to the particular business of preventing natural circulation.

The thermostats for energizing the pumps are located in strategic locations about 5 ft above the floor. Inside walls are preferred to outside walls as a place for mounting, because the latter may be quite cold and create a local coolness that is not related to the general room or building conditions.

In zoned buildings, a thermostat is needed for each zone.

A clock and thermostat combination offers a type of control particularly suited to buildings in which there is no night activity. The thermostat is set for a day temperature and for a night temperature; then the clock sees to it that each thermostat setting takes over at the proper time.

The combination of clock and thermostat is about double the price of the thermostat alone, certainly not a heavy investment for 24-hr automatic control.

Control of Water Temperature. The temperature of the water in the boiler may be kept at any value we desire. A thermostat with its bulb submerged in the water is the control. These thermostats are set to work over a range; thus if we decide we want water of 200 F maximum and 180 F minimum, the thermostat will start up the firing equipment at the lower temperature and shut it down when the water reaches 200 F.

This arrangement, being independent of the pump operation, ensures hot water for heating at any time, without notice. Up in the building the thermostat on the room wall orders the pump in the basement to furnish heat, the pump pulls the constant hot water out of the boiler and into the system, and everybody is pleased by the fast response.

The objective in operating the water thermostat over a range instead of at a single temperature is to avoid having the firing equipment worn out by what would be a continuous starting and stopping otherwise.

Indoor-outdoor Controls. Several systems are available which vary the water temperature in relation to the outside weather. Thus, in mild seasons, we may be kept quite comfortable with water in the radiators of 150 F rather than 40 or 50 deg higher. Naturally we save fuel and money at these lower temperatures, the saving resulting from the 40 to 50 Btu per lb of water that it is not necessary to add.

This type of control appears to have particular advantages when we use hot water for panel heating installations. For instance, if we have floor coils buried in concrete, heating the concrete takes time and so does the cooling. With an outside control anticipating the need for inside cooling or heating, we may not even be conscious of these lags. The heat is either turned off or on before persons inside the house are aware of a need for a change. Frequently it is all accomplished merely by raising or lowering the temperature of the water being circulated.

Miscellaneous Considerations

The Designed Temperature Drop. The temperature drop for which any system is designed is the difference in temperature between the water leaving the boiler and the water returning to the boiler.

General practice leans to a drop of 30 deg for a gravity system and of 20 deg for one with forced circulation.

The larger drop naturally creates a greater motivating force in gravity systems: this is a desirable condition. On the other hand, the larger the drop, the lower becomes the average temperature in the radiator, so that if we adopt a large drop we may also need to adopt more radiating surface. This costs money and is not a desirable condition.

As a practical example, let us consider a system with a 40-deg drop from 180 to 140 F, and then alternately a 30-deg drop from 180 F to 150 F. According to Table 4.1 the average water temperature of 160 F in the first case will provide a heat emission of 130 Btu per sq ft of radiation. By contrast, the 165 F water in the alternate case delivers 140 Btu, a clear increase of 10 Btu per sq ft. Obviously, we will require more radiation for the larger drop.

With forced circulation we can maintain the outlet temperature of a radiator fairly close to the inlet temperature, simply by pumping more water per minute to the radiator. However, we generally are satisfied with a drop of 20 deg because of the effect a lesser drop has on pump size. The more we pump, the bigger the pump, the bigger the pump, the bigger the cost, the bigger the cost, the louder the complaint from the man who pays the bill.

Occasionally, a designer will adopt a very large drop for a gravity system because the larger circulating head so provided enables him to use smaller pipe sizes. In these cases he hopes that the saving in piping costs will overbalance the added expense of more radiation and will show a little saving for the owner.

Adoption of such a plan must rest on careful judgment.

Air Venting. Wherever air collects in a hot water system we have a miniature compression tank. If the place of collection is in the piping, the air bubble is likely to obstruct the circulation and to give rise to complaints of poor heating.

As usual, prevention is simpler than the cure; thus in designing a hot water system we must be careful to do the following:

1. Arrange the piping so that, wherever a change in size takes place, eccentric fittings are used to maintain the top of the pipe in a straight line. Pockets where air can "hole up" are thereby avoided.

2. Connect upfed radiators at the bottom, on both inlet and outlet. Then air accumulating in the upper sections of the radiator can be vented without trouble, and circulation through the radiator is not checked.

3. Install automatic vent valves at high points of the system. Sometimes this can be accomplished simply by substituting an automatic air relief valve for the conventional plug vent on the highest radiator in the riser circuit.

4. Grade the piping in order to prevent air bubbles from finding pockets in which they will lurk on their uphill journey to the vent valves. This is not always practical because of low headroom, physical interferences, and zoning demands, but we will do the best we can.

Apart from the design, contractors and engineers should instruct the owner or system operator in the need for venting all radiators when the water is first heated at the start of a heating season. The summer's accumulation of air must go somewhere if the hot water is to fill the radiators, and the people who live with the system will make fewer service calls if they understand this fact.

Expansion and Contraction of Piping. We must always keep in mind that a piping system expands when warmed, contracts when cooled. This characteristic can embarrass us in tight quarters; on the other hand we use it to our advantage also.

The way to compensate for expansion in a tight spot is to use the pipe fitting trick called "cold springing." This means we will cut the pipe a little short, usually about one-half the expected distance of expansion, and then with a jack or block and fall we will force the reluctant ends to join. A circuit made up this way is under strain of course while cold, but as soon as the heat is turned on the expansion relieves the strain. As designers, we should indicate on our drawings the amount by which the connecting pipes should be cut short.

A hundred feet of iron or steel pipe may be expected to increase its length 1.2 in. when heated up from 70 to 200 F. Normally this expansion will give us no trouble with a hot water system, provided we do not tie down our circuits too tightly. The ordinary bends and changes of direction in elbows will easily compensate for so small a change in length. We need not even calculate the expansion in most jobs.

Systems going into tight quarters, however, require us to weigh the expansion problem with some care. The cause of a persistantly leaking joint may reside in the existence of an expansion stress, and no cure will be lasting until the stress is relieved.

Choice of Pipe. Whether we use wrought-iron, steel, or copper pipe is largely a matter of preference and price. Copper tubing delivered in large reels is handled almost like wire, so pliant is its nature; for this reason labor costs of installation will tend to be lower. As material, however, it costs more to begin with. Copper is less susceptible to corrosion, but with a hot water system, corrosion is less of a problem than with steam. Water fills the system at all times, and during the time when the temperature is less than boiling, gases are likely to remain in solution. Those which are released are soon vented in a correctly designed system. Table 5.8 shows dimensions of copper tubing.

TABLE 5.8. DIMENSIONS OF COPPER PIPE AND TUBING

Nominal pipe size, in.	Inside diameter, in. (pipe)	Inside diameter, in. (type L, tubing)
1/4	0.375	0.126
3/8	0.494	0.430
1/2	0.625	0.545
5/8	.....	0.666
3/4	0.822	0.785
1	1.062	1.025
1 1/4	1.368	1.265
1 1/2	1.600	1.505
2	2.062	1.985
2 1/2	2.500	2.465
3	3.062	2.945
4	4.000	3.905

Wherever there is a record of corrosion, we should obtain a water analysis in order to identify the corrosive agents. With this knowledge, we can introduce the proper protective measures, usually a matter of simple and inexpensive water treatment.

Wrought-iron pipe also has a record of high resistance to corrosion, thought to result from the rather chemically inert slag inclusions that are characteristic of wrought iron.

While ferrous piping generally is made up with screwed joints, copper tubing is soldered. For temperatures up to 250 F, which includes our hot water systems, a soft solder commonly is used. This solder may be of several formulas, one of them being 95 per cent tin and 5 per cent antimony and another 50 per cent tin and 50 per cent lead. For temperatures above 250 F, a silver solder is recommended. Silver solder also should be used, regardless of temperature, for all joints that are concealed, or that may be difficult to reach subsequent to erection. The reason is that silver soldered joints are as strong as the tubing itself, and we avoid a source of possible later trouble by making the joint as strong as we can. It costs a little more, but the assurance of satisfaction is worth it.

We should write our specifications accordingly.

Sizing the Hot Water Boiler. Having computed the heat losses from a structure, we are in position to convert these into a boiler for the job.

To see how this is done in practice, we now assume a few conditions. The load we will take as 150,000 Btu per hr and the water temperature at 195 F average in the radiators, and thus our radiators will emit 200 Btu/hr/ft^2. By dividing 150,000 by 200, we come up with 750 sq ft of radiation. We shall assume also that our boiler is fired by an automatic, atomizing oil burner.

Next, we shall examine a manufacturer's catalogue and see how closely an available boiler meets our requirements. Table 5.9 has been compiled from the data shown in a standard catalogue issued by a builder of heating boilers. All these ratings are based on the IBR code which we studied in detail in Chap. 4.

TABLE 5.9. BOILER SIZES FOR HOT WATER HEATING

Output, Btu/hr		Oil fired, gals/hr	Chimney		Net IBR radiation, sq ft*
IBR net	IBR gross		Size, in.	Height, ft	
94,000	140,000	1.50	8 × 8	30	625
114,000	170,000	1.80	8 × 8	30	760
135,000	200,000	2.10	8 × 12	35	895
155,000	230,000	2.35	8 × 12	35	1,030
175,000	260,000	2.65	8 × 12	35	1,170
196,000	290,000	2.90	8 × 10	40	1,305
217,000	320,000	3.15	8 × 12	40	1,450

* *Note:* Under the IBR rating code, the net radiation means actually installed radiation. Allowances are included by the builder in the boiler's capacity for pickup and piping requirements.

For our installation, we will not be safe if we choose the boiler rated at 760 sq ft, despite its apparent capacity for 10 sq ft of radiation more than we have. The reason is that the radiation (Table 5.9) is based on an emission of 150 Btu/hr/ft^2, whereas we expect to emit a third again that much per square foot, or 200 Btu. Therefore, we will go to the first column which deals with the net Btu output of the boiler and select the unit which is shown capable of delivering 155,000 Btu to the radiators.

This boiler will give us 775 sq ft of radiation in which each square foot is yielding 200 Btu per hr. This is 25 sq ft more than we need; thus we are now on the safe side.

The table shown has been prepared for the special purpose of illustrating how easy it is to get into trouble if we refer to a catalogue hurriedly and choose a unit before we understand all the conditions that govern the capacities published. Catalogue ratings always are modified by "ifs, ands, and buts," and ignorance will lead us not to bliss, but to disaster.

Other data of interest appearing in the table concern the chimney size and chimney height. Each manufacturer asks that we provide a chimney of certain dimensions, and if two manufacturers specify the same size for approximately equal boilers, we should consider this as coincidence only, and not as a rule. Boilers of the same capacity but of different styles often originate in the same factory—and their chimney requirements are different. One will need 40 ft of height while 30 may be enough for the other. Since chimneys cost money, the chimney column is worth our serious attention when we select a boiler.

To refer again to Table 5.9, the difference between the net and gross ratings in Btu per hour indicates the allowance available for meeting pickup and piping losses. If we have reason to believe the allowance is not enough for our particular conditions, then we should go to the next larger boiler. For instance, the manufacturer expects the piping to be covered. With hot water systems, covering is not mandatory except where outside exposure is encountered; consequently our piping losses will be higher than the boiler builder anticipated. If our piping is long or involved, we might be nonplused by the slowness of the radiators to warm up, despite our choice of a boiler that is *apparently* sufficiently large. In these cases, we probably have forgotten to allow a little extra pickup capacity to compensate for the bare piping. Correction then brings added expense in the form of covering for the piping.

Table 5.10. Maximum Capacity of Hot Water Piping in Equivalent Direct Radiation for 20-deg Temperature Drop*

Pipe size, in.	Friction per foot in milinches						
	10	50	80	120	160	240	300
½	15	30	42	54	64	83	87
¾	31	70	86	115	137	176	187
1	60	140	180	213	253	313	353
1¼	124	290	370	460	533	673	786
1½	186	453	553	700	826	1,033	1,163
2	364	862	1,063	1,320	1,562	1,913	2,146
2½	588	1,400	1,800	2,262	2,634	3,333	3,672
3	1,053	2,533	3,200	4,000	4,733	6,000	6,666
4	2,200	5,333	6,666	8,400	10,000	12,666	14,000

* *Note:* Equivalent direct radiation has been computed on the basis of 150 Btu per hour per square foot emission. The table may be used for other rates by applying the ratio $X/150$, where X = the chosen rate of emission.

Questions

1. Explain to your next door neighbor the fundamental differences between steam and hot water heating.

2. Describe, so that anyone can understand, how the motive force in a gravity system is created.

3. What is a milinch? A head of 5 ft is how many milinches?

4. Explain to a nontechnical friend the basic differences between the following systems:
   a. One-pipe, forced circulation.
   b. Two-pipe, reversed return.
   c. A two-pipe gravity, direct return.

5. Design a one-pipe forced circulation system for your own home.

6. In your own home, how many square feet of radiation will you need for average radiator temperatures of 200, 185, and 160 F?

7. Assume that the system you design for your own home is of the open type. What size expansion tank will you need? Where will you install it? For a closed system, what size compression tank will you need?

8. You are asked by an architect to discuss the heating for a nine-story building. What basic recommendations will you make? Why?

9. How will you safeguard your system against air pocketing? Itemize four steps to take in the design.

10. Select a boiler for your own house. How much radiation do you have on the IBR basis (150 Btu per sq ft emission)? How many Btu are available for pickup and piping? What size chimney do you need, and how high must it be?

CHAPTER 6

BOILER FIRING EQUIPMENT

Coal Firing Apparatus

The wide availability of coal as a fuel has inspired a multitude of inventors to design a multitude of ways in which to burn it. Basically, the successful designs may be shaken down into six classes of equipment, starting with the hand grates. After these come the power apparatus, comprising the traveling grates, the overfeed stokers, the underfeed stokers, the spreader stokers, and the pulverized-coal systems. We shall examine each in turn.

Hand-fired Grates. Probably most of us have shoveled coal on a heating boiler grate at one time or another. As with most firing equipment, these grates are cast iron, this material being able to withstand heat rather well. The most interesting feature of the hand-fired grate, from the designers point of view, is the rule for sizing. This rule is explained in Chap. 4, Boilers and Steam Generators, rather than in this chapter, because hand-fired grates in heating boilers are generally provided by the boiler manufacturer as an integral part of his equipment.

Traveling-grate Stokers. The stokers in this class are met in the larger boiler installations. As shown in Fig. 6.1, and 6.1*a*, a traveling-grate stoker receives the coal from a hopper at the front end of the boiler and thence carries the coal through the furnace to an ash pit at the rear. This sort of stoker actually forms a continuous moving belt of iron and steel, being driven by a sprocket at the front end, with an idler sprocket at the rear over which the grate reverses its direction and returns to the front, upside down.

Figure 6.1*a* is of particular interest because it shows the overhead furnace arch that is commonly used with traveling-grate stokers, although not always so pronounced as in this illustration. The purpose of the arch is to reflect intense heat downward on the green, incoming coal and to produce prompt ignition. There is a great deal of volatile in bituminous coal as conventionally used on these stokers, and if it is not swiftly ignited and burned as it rises from the coal, both smoking and a loss of efficiency result.

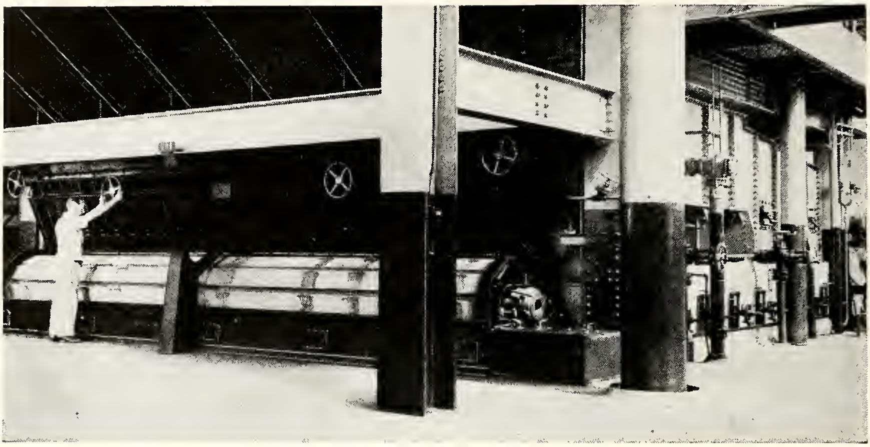


Fig. 6.1. Traveling-grate stoker, under a large boiler. Note the exposure of the grate links as they turn upward on the sprockets to reenter the boiler and receive a new charge of coal. (*Courtesy of Combustion-Engineering-Superheater, Inc.*)

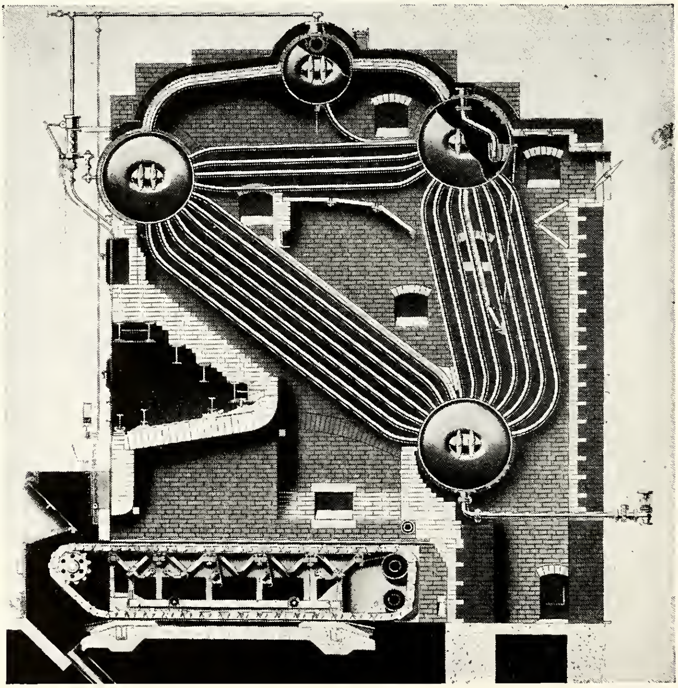


Fig. 6.1a. Traveling-grate stoker, side elevation, showing relationship of ignition arch. (*Courtesy of Illinois Stoker Company.*)

This arch is sometimes referred to as an "ignition" arch, obviously a rather apt expression.

Traveling grates have been considered by many engineers as being unsuited to the burning of anthracite coal, usually on the premise that the lack of volatiles delays ignition and may actually mitigate against a fire sufficient to cover the grate. However, anthracite has been burned satisfactorily in numerous plants with this type of equipment, even with water walls rapidly removing furnace heat. If anthracite is being considered, we should discuss the expectancy of satisfaction with the stoker builder before making a choice of design.

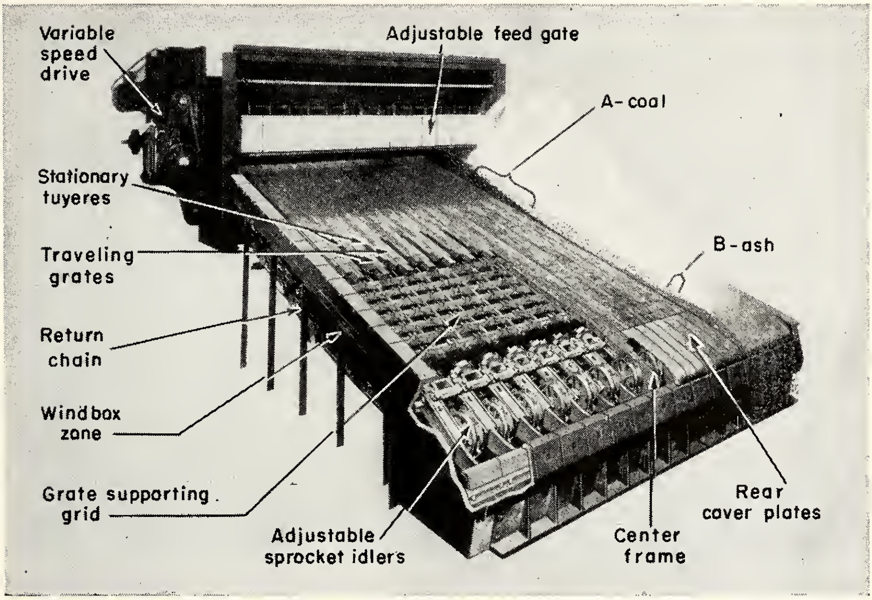


Fig. 6.2. The Stowe stoker, showing the alternately fixed and moving grate bars. (*Courtesy of Johnston and Jennings Co.*)

Traveling grates are susceptible to many variations in design, one of the most interesting being that shown in Fig. 6.2.

This is the Stowe stoker, in which alternate rows of grate bars are stationary and moving, the inventor's idea being that the stationary bars would set up a rotating motion in the coal as the moving bars carried the mass through the furnace and to the ash pit.

The use of preheated air and low ash coals is quite hard on stoker links and grates, as the preheating of the air eliminates what is otherwise a cooling bath on the iron, while scanty ash means that much of the grate will be exposed without adequate protection to the full heat of the furnace.

Overfeed Stokers. This form of coal firer is distinguished by it sharp angle of setting, often up to 45 deg, and the fact that the coal enters on top of the fire bed, and at the beginning of the slope. Gravity is put to work by these stokers, and helps the burning coal to tumble end over end in its descending journey to the ash pit. Just to make sure that the ash reaches the pit, a series of horizontal grate bars rise up and go down in regular sequence, pushing the coal down the slope by their actions.

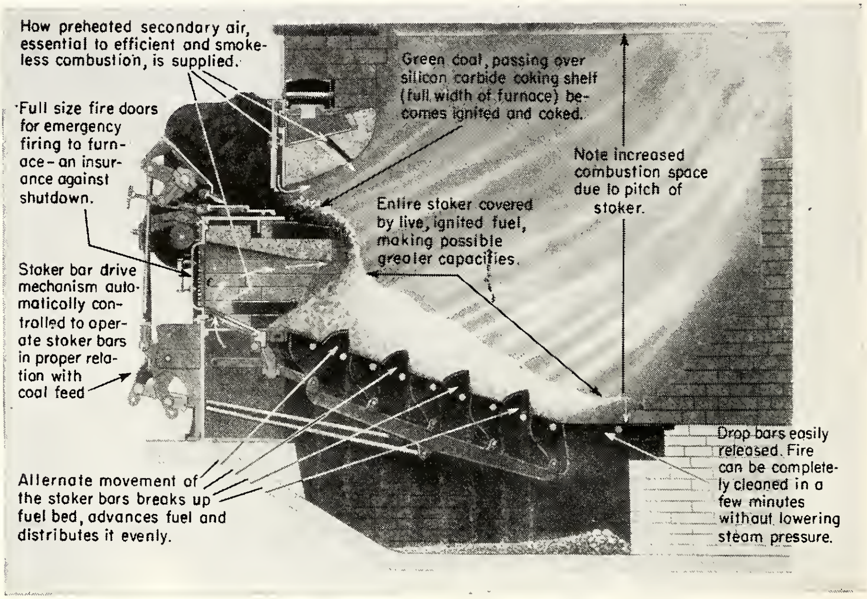

How preheated secondary air, essential to efficient and smokeless combustion, is supplied.
Full size fire doors for emergency firing to furnace – an insurance against shutdown.
Green coal, passing over silicon carbide coking shelf (full width of furnace) becomes ignited and coked.
Note increased combustion space due to pitch of stoker.
Entire stoker covered by live, ignited fuel, making possible greater capacities.
Stoker bar drive mechanism automatically controlled to operate stoker bars in proper relation with coal feed
Alternate movement of the stoker bars breaks up fuel bed, advances fuel and distributes it evenly.
Drop bars easily released. Fire can be completely cleaned in a few minutes without lowering steam pressure.

Fig. 6.3. The Huber overfeed stoker. (*Courtesy of Flynn and Emrich Company.*)

The grate generally slopes from the front to the rear of the furnace, with a variation appearing in the form of two grates, sloping from the sides to the center.

All the overfeed stokers employ an ignition arch, smaller usually than the arch for a traveling-grate stoker, but serving the same purpose. It ignites the volatile substances that are distilled from the green coal.

Overfeed stokers are used frequently for burning bagasse. Figure 6.3 illustrates a typical overfeed stoker.

Underfeed Stokers. The design of the underfeed stoker presents ideas of unusual interest, particularly when compared with the underlying objectives of the other designs.

Whereas the other stokers carry the coal straight through the furnace, or rotate it, or tumble it end over end, the underfeed design actually pushes the coal upward, and introduces fresh fuel to the flames from below the flaming surface. The mechanism for carrying out this plan consists of a ram, or series of rams, which alternately advance and retreat within a roofless, horizontal gutter, which is technically called a "retort." The coal enters the retorts at the front end of the furnace by gravity, from a hopper. Between each pair of retorts, there is a series of descending cast-iron steps, bearing perforations through which the air for combustion enters the bed of blazing coal. These steps are called "tuyères."

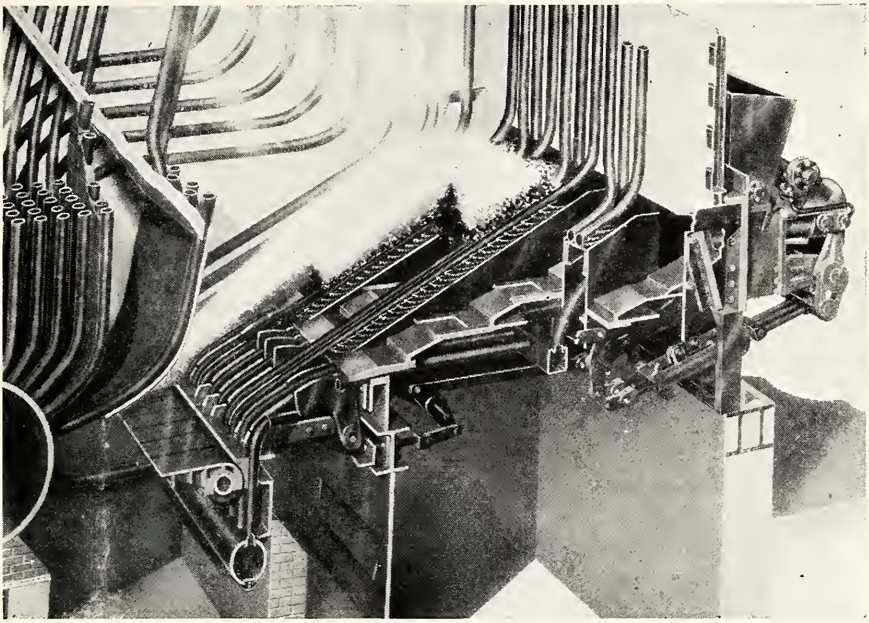


Fig. 6.4. Cutaway view of multiple retort, water-cooled, underfeed stoker. *(Courtesy of American Engineering Company.)*

Underfeed stokers have no need for an ignition arch, since the volatiles must actually pass through a ceiling of flames, as they rise from the incoming coal.

Underfeed stokers produce somewhat higher rates of firing than do the other designs, and consequently are found in many public utility stations where the load fluctuates both widely and swiftly. Some difficulties may be encountered on high firing rates with caking coals, since these coals tend to build up a solid cover of fused materials over the tuyères and effectively check combustion by starving the fire for air.

The problems presented by the use of preheated air are not any less urgent with the underfeed design, since overheated iron will burn in any arrangement. To meet and combat this situation, water cooling has been introduced into the design of large and heavily driven units. Figure 6.4 illustrates an underfeed stoker so cooled and protected.

While some proponents of the underfeed design claim for this stoker a greater smokelessness of operation than for the others, the fact is, an improper installation can smoke out the neighborhood, regardless of the stoker design selected. A common error of installation is seen when the boiler is set closely above the fuel bed. Little can be done to stop smoking in these cases, since the gases from the burning coal are so chilled by too early contact with the boiler surfaces that complete combustion is prevented, and smoke, soot, and cinders will both foul the boiler and emerge from the stack. If the setting cannot be changed, the introduction of air jets, under pressure into the furnace, is the best treatment.

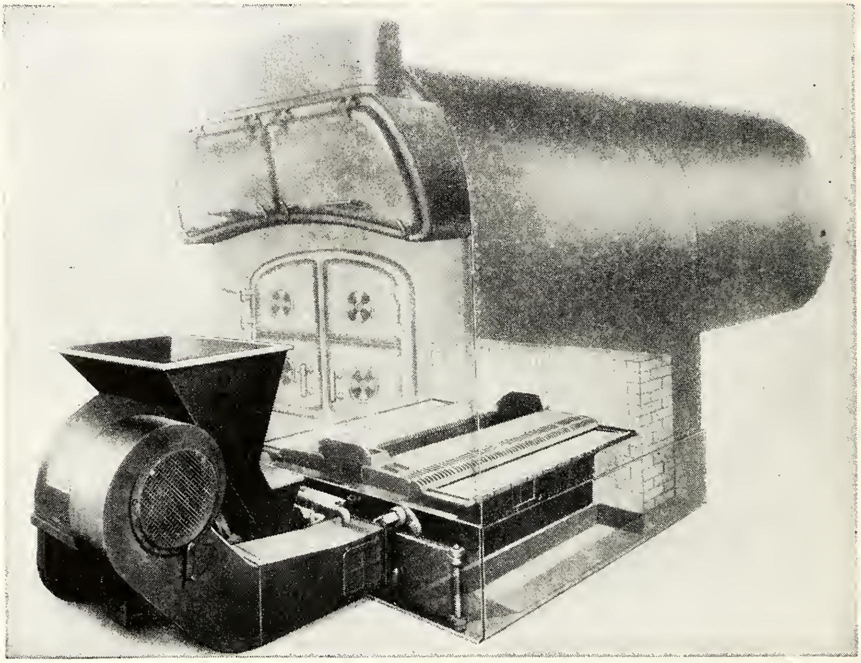


Fig. 6.5. Single retort underfeed stoker installed in a firebox type of bolier. ***(Courtesy of Detroit Stoker Company.)***

Figures 6.5 and 6.6 illustrate a type of single-retort underfeed stoker very common in industrial plants.

Spreader Stokers. The idea behind the spreader stoker is to bring each particle of coal into intimate contact with sufficient air for fast and efficient combustion. The spreader may be regarded as something of a link between the earlier stoker designs and the pulverized-coal systems.

Spreader variations again illustrate the fact that, given a basic idea, no two men will work out the details in the same way. Consequently, our spreaders may be purchased with either mechanical or pneumatic injection of the coal.

With the mechanical arrangement, the fuel feeds downward by gravity through a hopper, until it arrives on a feed plate. The feed plate is kept in motion, either rotary or reciprocating, and serves to deliver the coal to the blades of a revolving distributor; this part of the stoker suggests a paddle wheel.

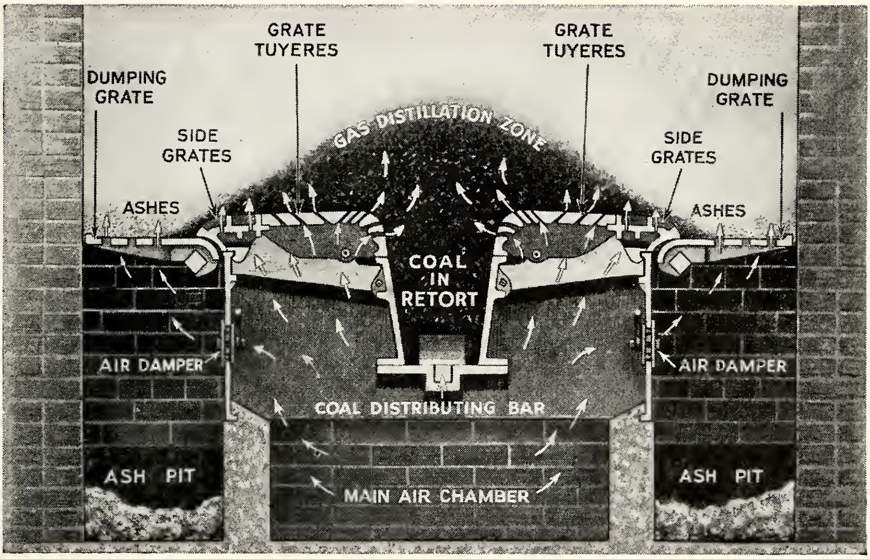


Fig. 6.6. End view of single retort underfeed stoker. (*Courtesy of Detroit Stoker Company.*)

The revolving distributor blades receive the horizontally moving coal from the feeder plate and heave it energetically into the furnace. As the fuel sails through the air, it is ignited, the fine particles burning in suspension while the larger pieces fall to the grate and complete their combustion there. To dispose of the ash, the grates are dumped from time to time, either manually or by power mechanism. In the smaller sizes, stationary grates may be used, the ash then being cleaned by hand.

The air for combustion may be introduced from air ports in the front wall between the fuel distributor and the grate or from a blast chamber underneath the grate or both. In times of necessity, natural draft may be used through the grate. This form of spreader is shown in Figs. 6.7 and 6.8.

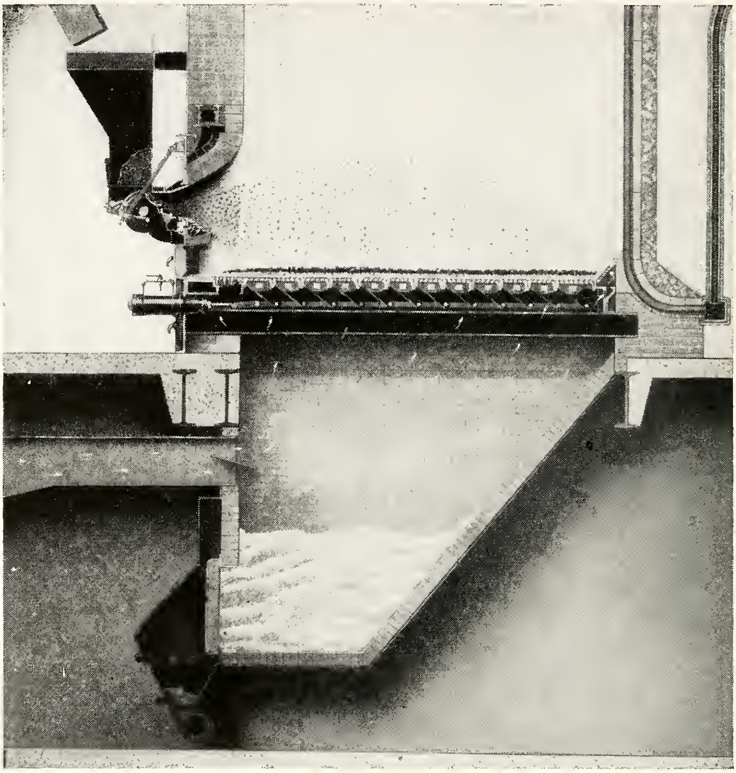


Fig. 6.7. Spreader type of stoker with power-operated dumping grates and ash pit. Stationary grates are provided if desired. (*Courtesy of Detroit Stoker Company.*)

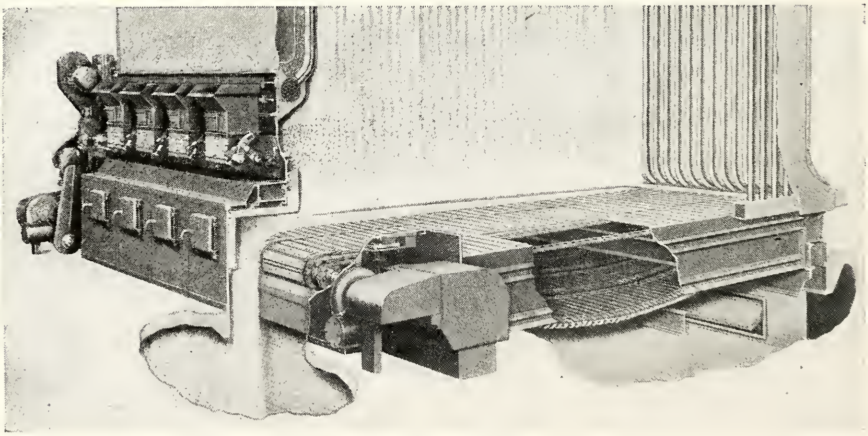


Fig. 6.8. Spreader type of stoker with traveling grate for continuous ash discharge. (*Courtesy of Detroit Stoker Company.*)

Figure 6.9 illustrates the arrangement of the pneumatic spreader. The coal is carried by a worm conveyor from the bottom of the hopper to a mixing chamber on the discharge side of the fan. Here the coal is picked up by the air blast and projected into the furnace from a nozzle.

Air to the fan may be abstracted from the furnace, which thus serves to preheat the coal, or the air may be taken from the boiler room, or the two temperatures of air may be mixed to provide whatever delivery temperature is desired.

As with the mechanical spreader, combustion air is provided from a plenum or blast chamber underneath the grate, and ash is removed either manually or by dumping the grates.

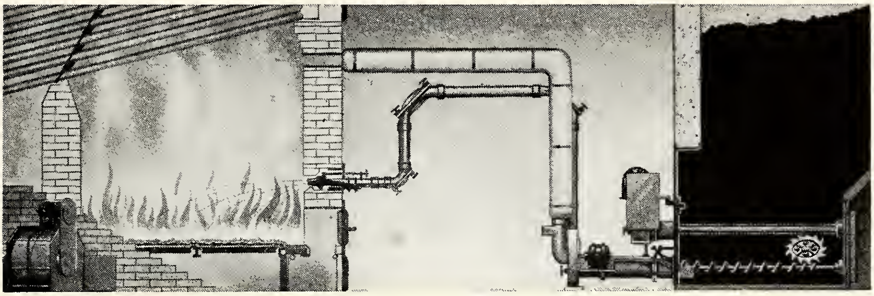


Fig. 6.9. Pneumatic type of spreader stoker. Note that motor-driven fan takes hot air from upper part of furnace to convey fresh coal to fire. (*Courtesy of Iron Fireman Manufacturing Company.*)

Spreader stokers are suitable for any size and type of boiler. They have no difficulty in developing 80 per cent or more combustion efficiency, which is comparable with other types of stokers.

We should be careful, however, in selecting a spreader stoker where fly ash could be a neighborhood nuisance. Sensitive areas indicate the need for some sort of fly-ash collector, either one of the commercial designs offered by various manufacturers or a system devised and built by the stoker manufacturer. In either case, common practice leads the recovered fly ash directly back to the furnace, where much of it is consumed.

Pulverized-fuel Systems. The idea of burning pulverized coal sold itself to industry because pulverizing made good fuel out of many hitherto poor grades of coal.

Early pulverized-coal designs frequently employed the bin system, which meant merely that the coal was pulverized more or less continually and stored in a bin against the moment of use. However, the building of these bins was expensive, and the incidence of fire, which imposed on the operators the task of digging out the burning areas, exercised its disagreeable influence when a decision was to be made; and consequently the current trend is to pulverize the coal as it is needed. This practice is referred to as the "unit system," with each boiler having its own pulverizer.

When we buy a pulverizer, we have a choice of designs, as usual. Figure 6.10 shows a mill designed to crush the coal by grinding it with rollers. In Fig. 6.11, the pulverizing is accomplished by a battery of hammers which are mounted suitably to a revolving shaft, and as the shaft turns, the hammers literally beat the coal to powder. A third type of mill accomplishes its purpose by simply revolving a conical drum in which the coal and cast-iron or alloy balls are mixed intimately. The pulverized coal is removed continuously by the air blast, which usually is well preheated.

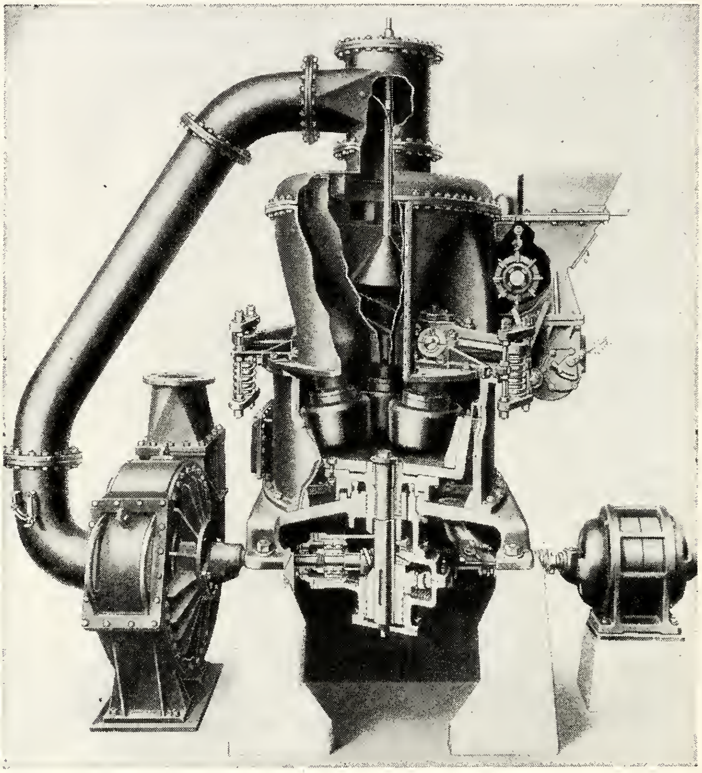


Fig. 6.10. Roller type of coal-pulverizing mill. (*Courtesy of Combustion Engineering-Superheater, Inc.*)

The principle of the ball mill, as this arrangement is called, has its prototype in nature. Birds, being toothless, are accustomed to swallowing small pebbles which bounce about in the craw as the bird moves, and thus mascerate fresh food until it is suitable for digestion. Figure 6.12 shows a ball mill.

Maintenance on pulverized-coal systems tends to be high, as many grades of coal suitable for pulverized firing are highly abrasive. Some of the mill manufacturers guarantee the maintenance of their equipment not to exceed a certain cost per ton of coal ground.

Pulverized-coal firing requires some form of fly ash and cinder catcher.

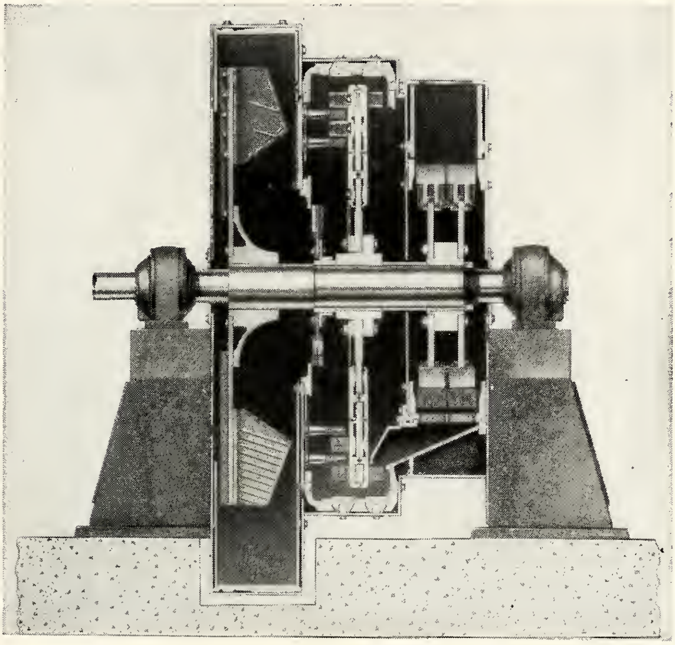


Fig. 6.11. Hammer type of coal-pulverizing mill. *(Courtesy of Riley Stoker Corporation.)*

Cinder Catchers. Three forms of cinder and fly-ash collectors currently used by industry are the following:

1. The change of direction trap is illustrated by Fig. 6.13. In this device the flue gases impact on baffles, turn corners, and otherwise deflect the air-borne particles into a collecting pocket or hopper by gravity.

2. Figure 6.14 illustrates the centrifugal principle. The entering flue gas is given a whirling motion by vanes or spiral ribbons, depending on the particular unit considered, and the ash being slightly heavier is thrown out of the stream. Centrifugal precipitators are sometimes called cyclone precipitators.

3. The third form is shown in Fig. 6.15; it separates the particles by means of electricity. In operation, the particles entering the precipitator

are given an electrical charge of definite polarity. Then they proceed to an area of opposite polarity, and since, electrically speaking, opposite poles attract each other, the moving motes promptly abandon the gas stream for the alluringly charged surfaces that have been placed in their pathway to the stack. Once on the plates, they stay, for the collecting surface has been made sticky, and their subsequent removal must be accomplished by means of mechanical scrapers or brushes.

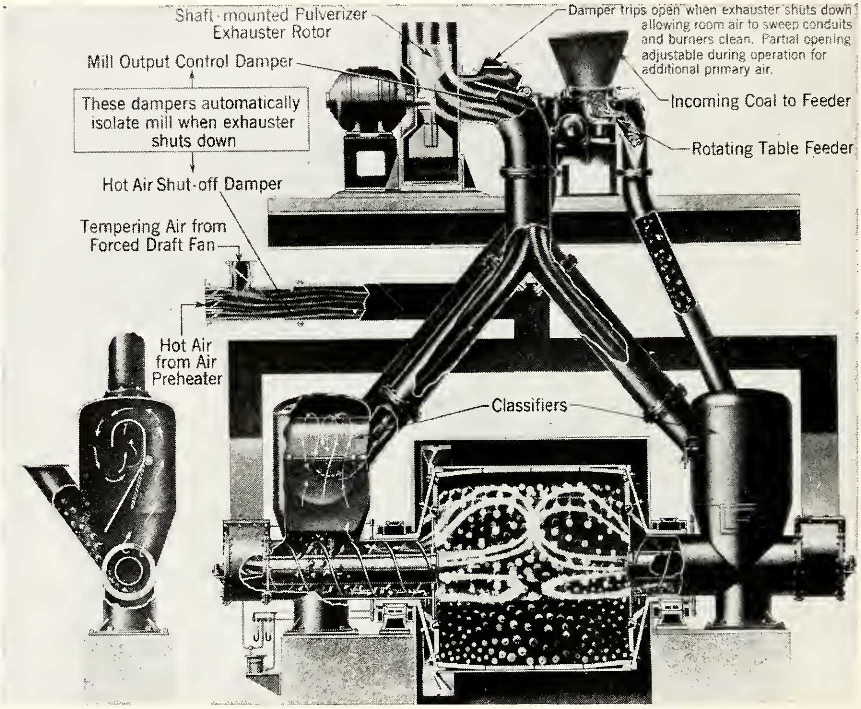

Shaft-mounted Pulverizer Exhauster Rotor
Damper trips open when exhauster shuts down allowing room air to sweep conduits and burners clean. Partial opening adjustable during operation for additional primary air.
Mill Output Control Damper
These dampers automatically isolate mill when exhauster shuts down
Incoming Coal to Feeder
Rotating Table Feeder
Hot Air Shut-off Damper
Tempering Air from Forced Draft Fan
Hot Air from Air Preheater
Classifiers

Fig. 6.12. Ball-mill type of coal pulverizer. (*Courtesy of Foster-Wheeler Corporation.*)

The critical size of fly ash and dust motes, at least from the commercial viewpoint, is 20 microns. A micron is 1/25,400 of an inch, and anything larger than 20/25,400 of an inch is considered likely by smoke-abatement engineers to become a nuisance. Our finest screen mesh has 325 openings to the square inch, and this screen will pass anything smaller than 43 microns, a situation that emphasizes the difficulty of cleaning our flue gases of these pin-point furnace outcasts.

Sizing Stokers and Cinder Catchers. The fundamental rule for choosing a stoker of any type is this: Select one that will burn enough coal per hour to satisfy the maximum demand for steam that is expected to be placed on the boiler.

The arithmetic of changing pounds of steam per hour into pounds of coal per hour is illustrated by the following procedure:

Suppose that we have a boiler turning out 100,000 lb of steam per hour at 140 psig, saturated, when supplied with feed water at 140 F.

The total heat in the steam at 140 psig saturated, from Table 1.1, is 1193.0 Btu per lb.

The heat in the feed water is 140 F − 32 F = 108 Btu per lb.

The difference between 1193 and 108 Btu represents the quantity of heat that must be added to each pound of water to convert the water to the desired steam; it is 1085 Btu.

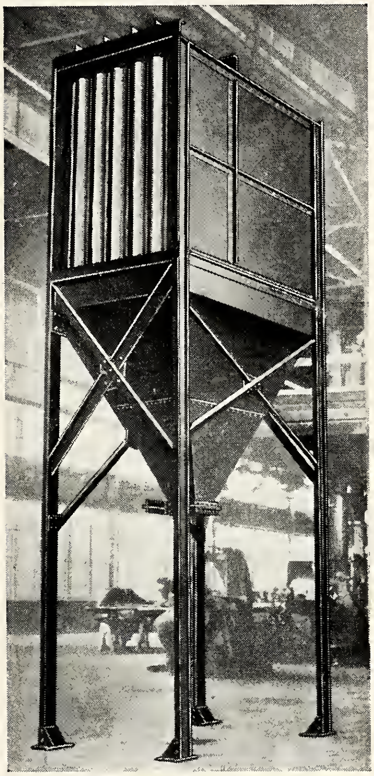


Fig. 6.13. Impact type of cinder trap. (*Courtesy of Green Fuel Economizer Company.*)

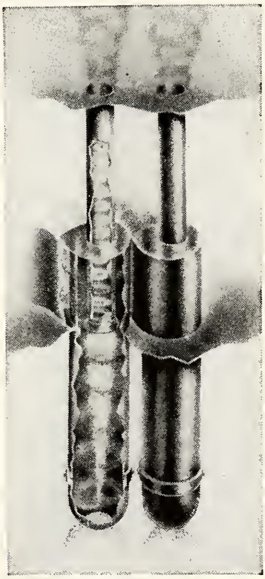


Fig. 6.14. Cyclone type of dust precipitator. (*Courtesy of Prat-Daniel Corporation.*)

The total heat output of the boiler in useful steam is therefore 100,000 lb multiplied by 1085 Btu per lb, or 108,500,000 Btu per hr. If the over-all efficiency of the boiler and stoker is 80 per cent, then obviously the total heat input to the boiler must be 108,500,000/0.80, or 135,625,000 Btu per hr.

Next we divide this total input by the calorific value of the coal *as fired* on the stoker, and the result is the number of pounds of coal per hour that the stoker must burn. Assuming in this case that our coal has a calorific value as fired of 13,200 Btu per lb, we must select a stoker capable of burning at least 10,275 lb in any one hour.

Sometimes a boiler output will be given in terms of boiler horsepower, with the boiler operating at an increased rating. For example, we may buy a 200-hp boiler to operate at 300 per cent of rating. The total output then becomes 200 × 3, or an actual 600 boiler hp at the peak load.

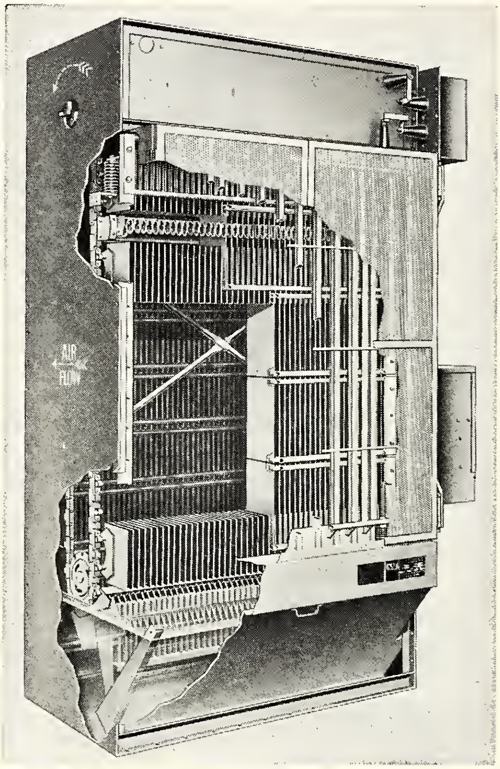


Fig. 6.15. Electronic self-cleaning air filter. (*Courtesy of American Air Filters Company.*)

The firing equipment is then sized on the equivalent heat input for 600 boiler hp.

When superheat is involved, this factor is incorporated in the total heat of the steam. For instance, if we had say 27 deg of superheat in our 140 psig steam, the total temperature of the superheated steam is 353 F (saturated temperature) plus 27 F or 380 F, and the total heat becomes 1209.4 Btu per lb.

For sizing the cinder catcher, we start with the pounds of coal per hour that we expect to burn at maximum load. The manufacturer of the firing equipment who already has given us the over-all efficiency of the boiler when operating over his machinery also provides information on flue-gas temperatures at the boiler outlet.

Now we approach the cinder-trap builder and tell him the following things:

1. The number of pounds of fuel we expect to burn in one hour.

2. The nature of the fuel, volatile percentage, ash content, and so on, An analysis, either proximate or ultimate, will be helpful.

3. The temperature of the flue gas at the boiler outlet.

4. The over-all efficiency of the steam generator and firing equipment, the per cent of CO_2 or any other data we have that illuminates the condition of the flue gas and its constituents.

5. The efficiency of fly ash and cinder recovery that we desire. Guarantees up to 98 per cent may be obtained where the particles are 20 microns in size, and larger. For particles smaller than 20 microns, up to 85 per cent collection is possible, depending of course on the actual fineness encountered.

The manufacturer of the precipitator will then size the unit to meet our needs. Again, as with stokers, the maximum load is the critical load. Precipitators sized for 200 boiler horsepower cannot be expected to serve efficiently if the boiler output is run up to several times this value.

While the application of cinder catchers to simple heating boilers is not generally necessary, some cities are requiring this equipment under smoke ordinances for the larger installations. Apartment houses, for example, which operate large and sometimes high-pressure boilers may expect to come under a nuisance scrutiny, and in one city at least, mechanical collection is mandatory for building incinerators.

Oil Burners

Basic Principle of Design. The purpose of the oil burner is to break up the fuel into such minute particles that instant and efficient mixing of the oil and combustion air is achieved. To gain this desirable end, we have a choice of four methods, each of which we shall examine in turn.

Steam-jet Atomizers. Oil is delivered to the burner barrel at low or medium pressure, and is then carried into the furnace on a jet of steam which simultaneously shatters the oil into a fine spray or mist.

Steam-jet atomizing burners are simple in design, have considerable flexibility on load fluctuations, and can be used successfully with heavy oils, and even for atomizing tars. On the debit side, they consume up to 3 per cent of the total steam produced by the boiler, depending of course on the boiler load and on the density of the fuel. Steam-jet equipment is not suitable for marine service, since the atomizing steam is a complete loss up the stack, and the resulting make-up imposes an unacceptable burden on the fresh-water facilities of the ship.

A steam pressure of at least 100 psig generally is needed for satisfactory atomization, although 25 psig may work with some burners.

Figure 6.16 illustrates the general arrangement of the steam-jet atomizing burner.

Air Atomizing Burners. Burners designed for steam atomization may be used for high-pressure air, if desired. Finding a source of compressed air is not always done easily, and can add substantially to the expense of the installation. We may be obliged in fact to buy an air compressor with all the auxiliary equipment that goes with such a system.

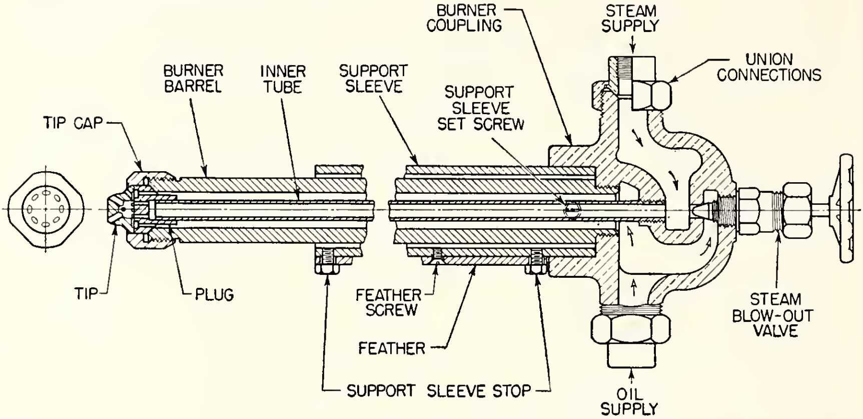


Fig. 6.16. Steam-jet type of atomizing oil burner. (*Courtesy of Peabody Engineering Corp.*)

Most of the modern air atomizing burners use low-pressure air, operating on the premise that a large volume of low-pressure air actually delivers as much energy as a reduced volume of high-pressure air. For instance, Kent reports that 52 per cent of the total combustion air is required for atomizing at an air pressure of ½ psig, whereas only 7 per cent is ample at 100 psig.

Many of the modern package type of steam generators operate with low-pressure air atomization, the capacity of the forced draft fan being easily able to meet the volumetric needs.

Low-pressure air functions best on light oils. With Nos. 5 and 6 oil, extensive preheating of the oil is needed if the burner is to perform satisfactorily.

Mechanical Pressure Atomizing Burners. The key to successful mechanical atomization is found in a happy marriage of adequate preheating and high delivery pressure.

The temperature should be high enough to provide a viscosity at the burner of 20 sec Saybolt Furol, or 150 sec Saybolt Universal. With the heavy No. 6 oils that long carried the name of "bunker C," this meant a temperature of approximately 220 F.

Pressures needed vary with the design of the burner, but a conventional range of 200 to 300 psig may be expected.

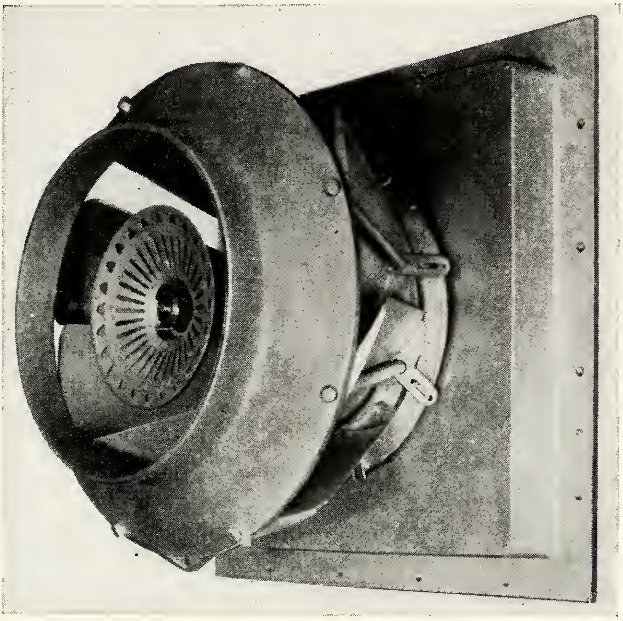


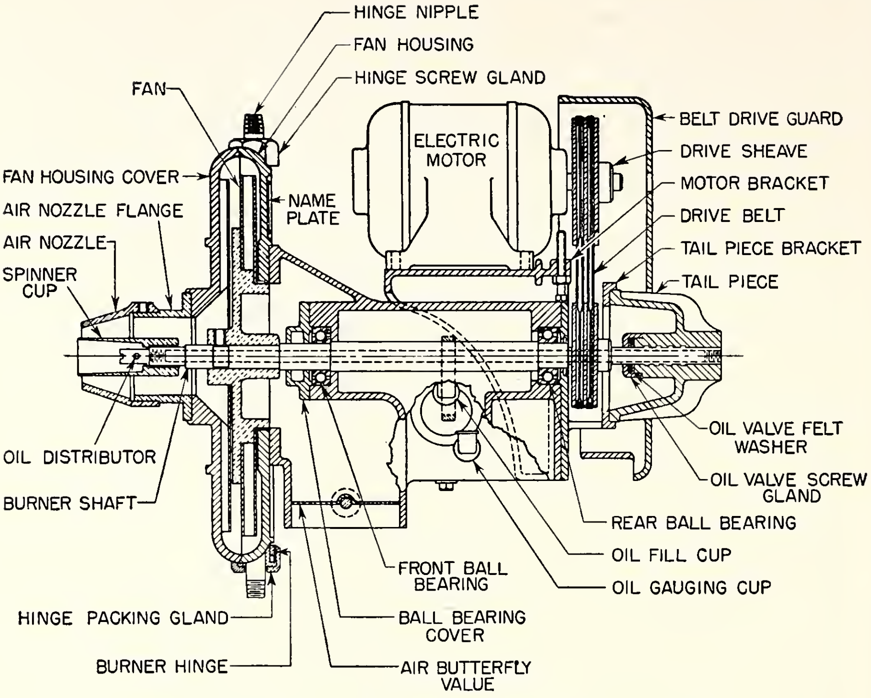
Fig. 6.17. Pressure type of atomizing nozzle. (*Courtesy of Peabody Engineering Corporation.*)

The pressure atomizer is designed with a sprayer plate at the furnace end of the delivery pipe, this plate usually incorporating three or more tangential grooves, or slots, through which the oil is forced at right angles to its previously horizontal line of flow, converging to the center of the plate, whence it issues through an orifice in the form of a fine spray. The tangential slots impart a whirling motion to the oil, thus causing the spray cone on the furnace side of the orifice to be thin-walled and cyclonic, two conditions that contribute happily to the mixing of the oil with the air in the furnace.

Combustion air for mechanical pressure atomizing burners usually enters the furnace from an air cone, or ring, mounted on the boiler front and circumscribing the burner.

Pressure atomizing burners are common in marine service and are standard with the Navy.

Figure 6.17 illustrates the arrangement of a pressure atomizing burner.



NOTE: WHEN ORDERING PARTS SPECIFY SIZE OF BURNER, PART NUMBER, SERIAL NUMBER, CURRENT CHARACTERISTIC, AND OTHER DETAILS TO FULLY DESCRIBE ITEMS REQUIRED

Fig. 6.18. Rotary type of oil atomizing burner. (*Courtesy of Cleaver-Brooks Company.*)

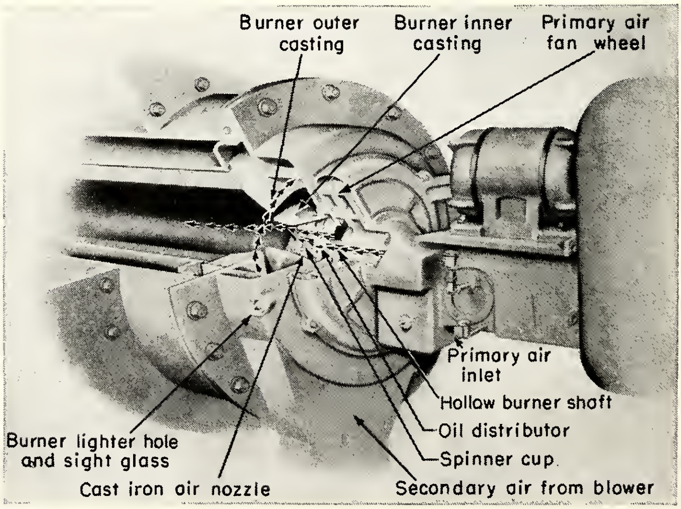


Fig. 6.18a. Rotary burner in package steam generator. (*Courtesy of Cleaver-Brooks Company.*)

Rotary-cup Burners. In this design, the oil is pumped at moderate pressure, perhaps up to 50 psig, into the burner barrel through which it travels to a truncated cone, or cup, with the larger end of the cup open to the furnace. This cup rotates at the same speed as the motor, generally about 3,450 rpm, and the oil which is fed into it is whirled off the lip at the open end, to form a hollow, thin-walled cone.

The taper of the cup walls affects the shape of the flame. A wide taper produces a short flame; narrow tapers, long flames. The shape of this flame with all burners should fit smoothly the shape of the combustion chamber.

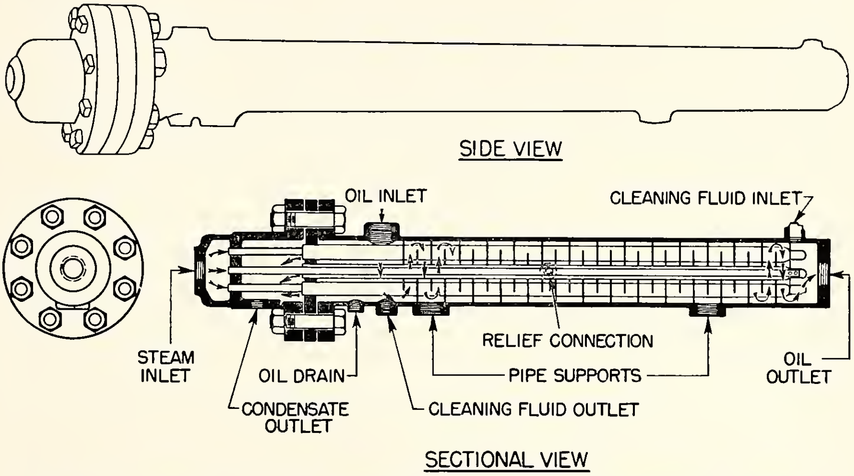


Fig. 6.19. Steam type of oil preheater. (*Courtesy of Cleaver-Brooks Company.*)

Primary air for combustion is introduced from an air nozzle arranged in annular form around the lip of the cup, and for best results, retracted about ⅛-in. from the lip. If the air ring projects too far, it will blow the flame away from the cup. In any event, the supply of air should be controlled by a damper in the blast pipe.

Rotary-cup burners appear to be gaining popularity, their ability to burn any grade of oil being a distinct asset. They can handle oil as heavy as No. 5, without preheating.

Figure 6.18 shows a typical rotary type of atomizing burner.

Preheaters. An oil preheater is simply a heat exchanger. In the steam type illustrated by Fig. 6.19, we observe a bundle of steam-filled tubes totally submerged in oil. Every couple of inches, the tubes pass through a circumferential baffle, which alternately directs the journeying oil up and down and from side to side, across the tubes. By the time it escapes from this contrary maze, the oil is adequately heated. The condensed steam is drawn off through a trap and returned to the feed-water system. Fig. 6.20 shows a temperature regulator.

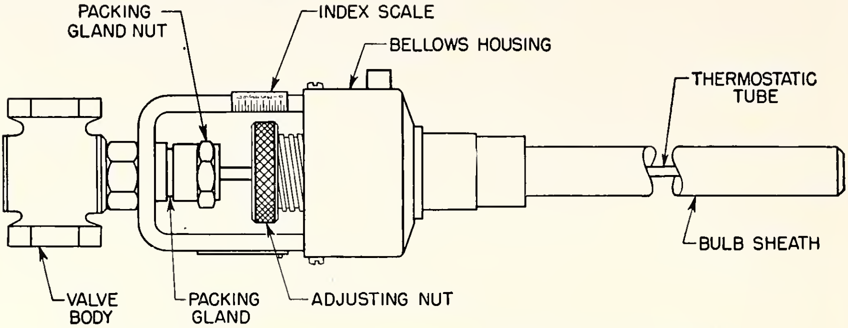


Fig. 6.20. Fuel-oil temperature regulator for steam preheater.

There are times in every installation when steam for heating the oil is not available, for example, when starting a boiler that has been shut down for several days. To meet this contingency, we must provide an electric fuel-oil heater, as shown in Fig. 6.21.

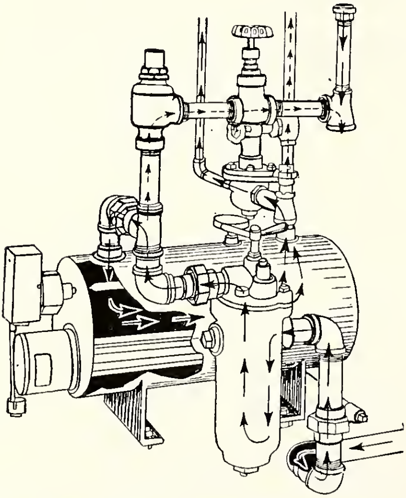


Fig. 6.21. Electric type of oil preheater. (*Courtesy of Cleaver-Brooks Company.*)

Electric heaters should be cleaned from time to time, since the oil will tend to scale the electric heating element and thereby both reduce its capacity and create some danger to the element from overheating.

In the event we have neither electric heating equipment nor steam, then we must start the boiler with an auxiliary pump, perhaps driven by a gasoline engine, and use a light oil which is not affected seriously by cold weather and atomizes easily without preheating.

Sizing the Oil Burner. In most of the residential and small commercial heating problems that we encounter, the boiler we select will incorporate an oil burner as a matter of course. No question of size demands an answer from us in these cases.

For the larger installations, in which we stipulate the capacity and type of burner we intend to buy, the sizing procedure follows the stoker-sizing pattern, with one exception. Instead of using the calorific value of coal, we substitute the value of oil. The pounds of fuel oil burned per pound of steam produced will be less than with coal, for the calorific value of the oil is greater.

Ignition and Safety Devices. When oil vapors are mixed with air, the result is an explosive mixture. This fact of nature is the cause of numerous puffs and flarebacks that occur in oil-fired furnaces.

The dangerous moments occur between the times of burner starting and ignition. If ignition is delayed for many seconds, the volume of explosive vapors building up in the furnace may contain sufficient potential energy to blow all doors off the boiler and to destroy the brick setting. For this reason, all burners are built with ignition monitors that stop the burner automatically if the oil fails to light off within a safe interval. The same monitors prevent the burner from attempting a restart until natural draft has vented the furnace, a period of from 1 to 3 min.

Despite the presence of all the safeguards that engineering ingenuity has so far devised, explosions do happen, and for this reason there is one basic safety rule that all of us should observe: When present at the lighting of an oil burner, never stand in direct line with any furnace door or opening. We may disregard this rule a thousand times, and there may be no puff; on the very next occasion we may lose our eyebrows at best, perhaps spend a period in the hospital. Men have been killed in this very circumstance.

A second rule worth remembering, especially since it applies to the smaller residential and commercial furnaces, is this: Always inspect the furnace for oil pools before lighting off, and never attempt to start the burner until all pools, if present, are wiped up. At the same time, provide all the draft possible through the furnace.

Lighting large industrial boilers, say from 500 hp up, is accomplished usually with a torch. We can fabricate a successful torch from ¼-in. steel rod about 4 or 5 ft long. One end is bent to form a handle, and at the other end, 5 in. apart, we weld a couple of 2-in. disks or washers which act as stoppers for a strand of asbestos rope wrapped around the rod between them. Rags or waste will serve if asbestos is not available, although the former is to be preferred.

To store a torch of this kind, we can make a pot of 6-in. pipe, 18 in. long, and close the bottom with a flat plate a foot square. If we keep plenty of kerosene in the torch pot, so that the level is above the asbestos on the rod, we can safely extinguish the torch after use by dousing it straight into the pot.

For a gas torch, a 5-ft length of ¼-in. pipe attached by a rubber hose to a valved gas line is quite satisfactory.

Many of these large boilers are fired with multiple burners, which can be lighted in sequence from the central burner, after that key flame has been established.

Two forms of ignition are common to the small-and medium-sized installations. Up to about 50 hp, an electric spark is conventional practice. From 50 to 150 hp, dual electric ignition may be used, one electrode being located on each side of the burner.

The larger boilers, and many of the small ones also, are now using a gas-electric combination system, in which a gas flame is ignited by electric spark. The gas flame then serves as a large torch for the main oil burner, the physical size of the flame being sufficient to reduce the likelihood of delay in the lighting of the oil.

City gas may be used in this system when available; bottled gas in locations distant from the city mains.

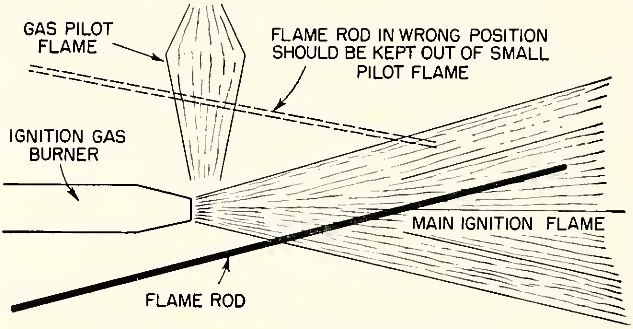

GAS PILOT FLAME
FLAME ROD IN WRONG POSITION SHOULD BE KEPT OUT OF SMALL PILOT FLAME
IGNITION GAS BURNER
MAIN IGNITION FLAME
FLAME ROD

Fig. 6.22. Arrangement of the flame rod in oil-fired furnaces.

All the larger burners start on what is called "low fire." Low fire means simply that the burner is feeding into the furnace a reduced quantity of oil, with the objective of making ignition less violent. Once ignition has been established, the oil valve opens automatically, and the burner assumes the state of "high fire" operation.

As a check on the ignition of the gas torch, we have what is termed a flame rod. As shown in Fig. 6.22, the flame rod extends into the gas flame, and if the gas is lighted by the spark igniter, a small current of electricity passes through the flame and into the flame rod, whence it journeys to an amplifier and is there stepped up in value until its strength is enough to start the main oil burner.

The electric eye, which monitors the main oil-burner flame, is not dependable where the gas flame is concerned, for its sensitivity to the bluish light of the gas flame is very low. The electric eye functions best with orange and yellow light, and will stop the burner immediately if the main oil flame is extinguished, once ignition has been established.

Figure 6.23 shows a typical photocell of the type used for furnace monitoring. So long as light of the proper wave length falls on the cell, a current is generated that maintains the oil valve in the open position. If the oil flame fails, the current is interrupted, thereby closing the valve.

Electric eyes are not affected either by incandescent furnace walls or glowing carbon, since the light emitted by these sources is of a long wave length to which the cell is not sensitive. The unit appears complete in Fig. 6.24.

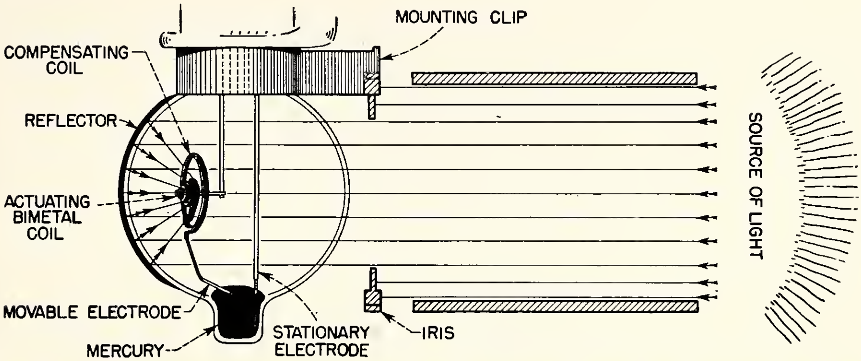


Fig. 6.23. Diagrammatic arrangement of the Visaflame electric eye, showing how the light waves actuate the movable electrode in and out of the mercury. (*Courtesy of the Mercoid Corporation.*)

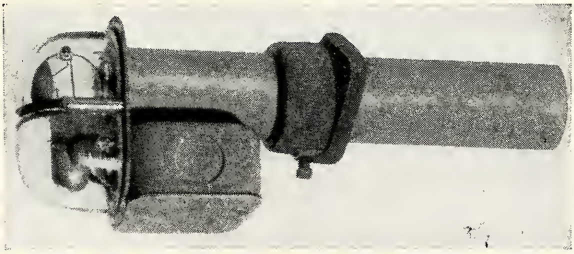


Fig. 6.24. Visaflame unit and sight tube that focuses light on sensitive element. (*Courtesy of the Mercoid Corporation.*)

Vaporizing Burners

Basic Principles. Instead of atomizing the oil, a vaporizing burner prepares the fuel for combustion solely by means of heat. The gases which arise from hot oil and which are potential explosion hazards in the mechanical burner systems are consumed under the vaporizing arrangement as rapidly as they form.

To illustrate the design, Fig. 6.25 shows a conventional pot burner, found wherever space heaters and floor heaters are used. The oil enters the furnace, or pot, by gravity through a float valve, and spills out over the floor of the pot. As a shallow pool forms, combustion can be started simply by dropping a lighted match to the surface of the oil. The tiny heat of the match vaporizes some of the oil which promptly ignites, thereby creating more heat which creates more vapors, which burn and create more heat to create more vapors, and so on, ad infinitum, or until the oil gives out. The rate of combustion is controlled by the float valve which admits oil to the pot on the basis of either manual or thermostatic control.

Vaporizing burners are sharply limited in the grades of fuel they can burn. Since it must vaporize easily, only No. 1 or No. 2 fuel oil is suitable for most of these burners. Kerosene will burn also, but is more expensive.

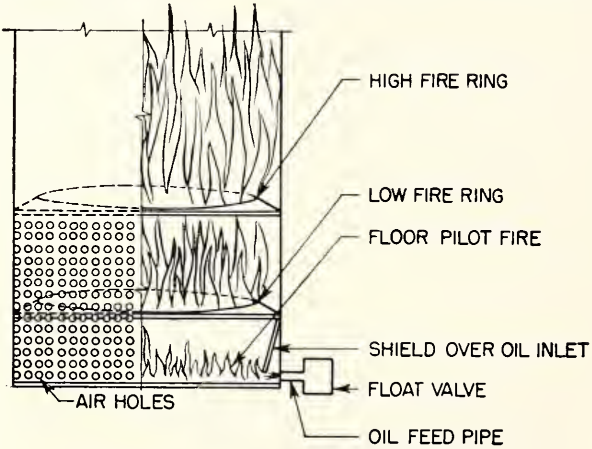


Fig. 6.25. Common type of vaporizing burner.

Vaporizing Design. If the oil feed "cracks" instead of vaporizing as the result of high temperatures in the pot, this situation is indicated by the formation of soot and hard carbon on the inner surfaces of the apparatus. To avoid this condition, the pot designer includes a baffle ring set several inches above the floor, thereby shielding the incoming oil from the direct radiant heat of the fire.

Generally, combustion takes place from the top of this baffle ring, or even higher at the upper fixed ring, depending on the rate of firing. The critical temperature is approximately 650 F; above this temperature, cracking of the oil takes place.

The bottom baffle ring never should be removed except for cleaning. If we leave it out, cracking is inevitable.

Persistent smoking and sooting in this type of burner is a symptom of air restrictions, more often than otherwise resulting from deposits of carbon. The pot should be thoroughly cleaned when these conditions are observed, the air holes being reopened with a wire brush, or a solvent such as carbon tetrachloride. The filter in the float valve also should be cleaned at regular intervals.

Gas Burners

Basic Principles. A properly designed gas burner performs the same function of mixing fuel and air that we observed to be the primary purpose of the oil burners. However, with gas fuel, the problem is strictly one of mixing, whereas the oil burners are required to shatter the liquid into a vapor or mist, as well.

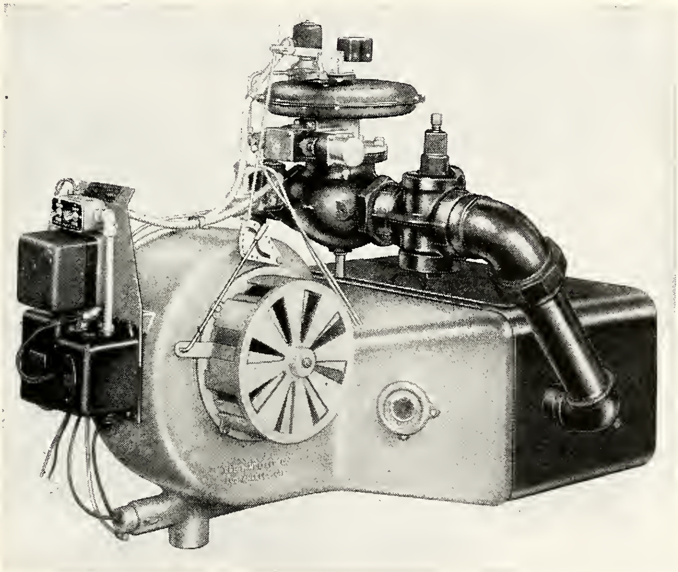


Fig. 6.26. Forced-draft gas burner, with integral fan. (*Courtesy of Lee B. Mettler Co.*)

The mixing is accomplished on the Bunsen burner principle.

Gas emerges from a tube under some positive pressure and with considerable velocity, thereby aspirating room air through ports which are set a little lower than the gas nozzle. With manufactured gas, 3 parts of air are aspirated for 1 part of gas; with natural gas, the proportion of air is almost double.

Because of their different mixing characteristics, burners must be changed when the gas is changed. Thus a burner designed for manufactured gas is not suitable for natural gas, or for liquefied petroleum gas.

Gas firing, as with oil firing, requires safety watchmen in the form of mechanical devices to perform continuous duty. For example, an inter-

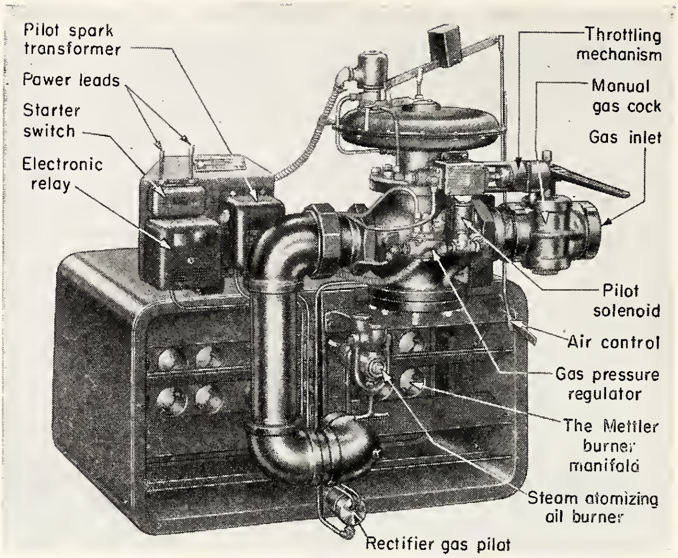


Fig. 6.26a. Natural-draft gas burner. (*Courtesy of Lee B. Mettler Co.*)

ruption to gas flow may extinguish a pilot light, and then, when gas service is restored, an explosive cloud is soon formed which will detonate at the first spark.

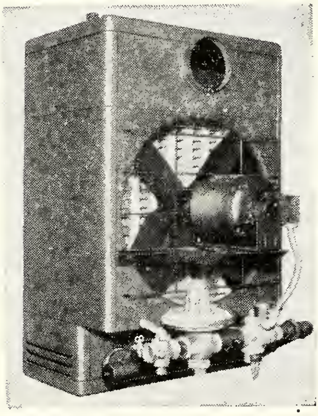


Fig. 6.27. Arrangement of gas burner and valves under unit heater. (*Courtesy of Automatic Gas Equipment Co.*)

Gas-burner Applications. Gas fuel is convenient to handle and of high calorific value; consequently it is a popular fuel for an entire range of applications, from large power boilers down to room-size, open-fire space heaters.

Figures 6.26 and 6.26a illustrate types of gas burners used for large boilers.

Figure 6.27 shows an installation under a unit heater, together with the necessary auxiliary valves and controls. Gas enters the combustion system through a manual control cock, passes through a pressure regulator, and then in turn is monitored by a solenoid valve which closes under thermocouple control if the pilot light fails. A second solenoid valve is actuated by the room thermostat.

If the two solenoids permit its passing, the gas next enters a manifold from which it is sorted out to the main burners.

Figure 6.28 shows an arrangement of gas piping that is recommended for large boilers having multiple burner fronts and is intended to prevent lighting off explosions due to the accumulation of gas from leaky valves.

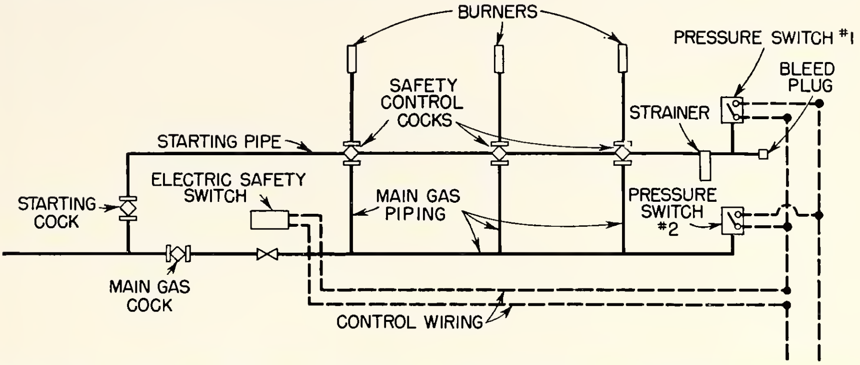


Fig. 6.28. Gas piping on boiler front designed to avoid explosions from leaking burner cocks.

In past years, this source of explosions has been our most prolific cause of destruction and tragedy in gas-fired industrial boilers.

Sizing the Gas Burner. Gas is metered in cubic feet, and its calorific value is determined on the same basis.

The standard combustion efficiency of the American Gas Association

Table 6.1. Capacities of Conversion Gas Burners*

Sq ft of installed radiation, steam	Gas pressure, oz	Estimated gas consumption, cu ft/hr	
		Natural	Artificial
400	½	175	350
600	1¾	250	500
800	3	350	720
1,000	1	420	840
1,600	2¾	650	1,300
2,000	1¼	780	1,560
3,000	2½	1,130	2,260
4,000	1½	1,500	3,000
5,000	2	1,880	3,760

* *Note:* Gas consumption, assuming 80% efficiency, depends on the calorific value of the gas at the point of consumption. City gas values differ from city to city, being based on local ordinance, while natural gas varies with its field of origin. For data on these gases, as well as for liquefied petroleum gases, see Chap. 14 on Fuels and Fuel Handling.

is 80 per cent; consequently we compute our Btu input to the furnace, boiler, or heater on this basis, and then divide the input by the calorific value per cubic foot of the gas we plan to burn. The answer of course is the number of cubic feet per hour we must provide.

For most of the gas-fired heating plants that designing engineers generally consider, the burners are part of the unit and present no problem. For industrial installations, the sizing should be referred to the burner manufacturer.

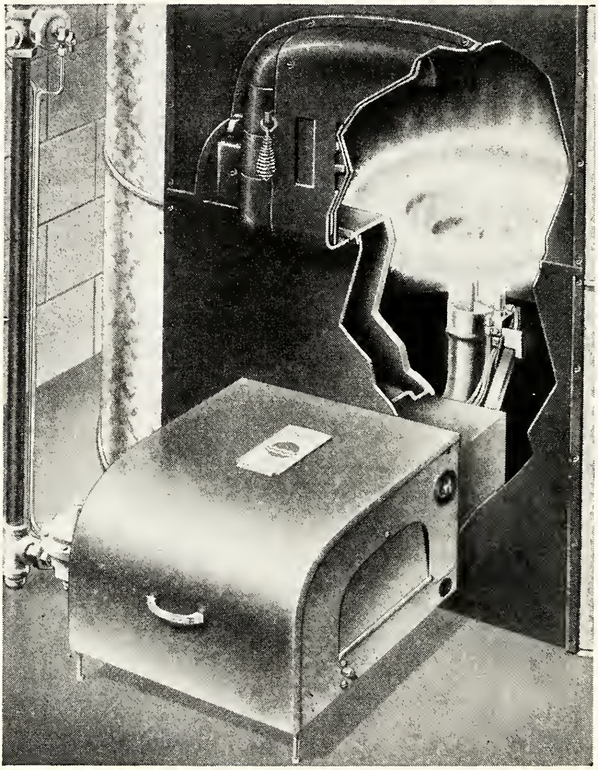


Fig. 6.29. Gas conversion burner as manufactured by the American Radiator and Standard Sanitary Corporation.

Conversion Burners. From time to time we encounter the problem of changing over a coal-fired, or oil-fired, boiler to gas fuel. The gas burner for such an installation is available from several manufacturers, and each builder supplies his appropriate instructions for the burner's lodgment. As an indication of the capacities of conversion burners, Table 6.1 is provided on page 153.

Questions

1. Name and describe four types of stokers currently in wide use.
2. What is an ignition arch? Where is it used?
3. Why is the spreader stoker so named? Describe two types of spreaders.
4. What is the advantage of pulverized coal?
5. Describe the three types of pulverizing mills presently being sold.
6. Describe three forms of cinder catchers. What is the basic collection principle illustrated by each?
7. What is a micron? What is the critical size of chimney particles from the nuisance viewpoint?
8. How large a stoker is needed for a boiler output of 50,000 lb of steam an hour at 150 psig saturated, and from a feed-water temperature of 130 F? Assume 75 per cent over-all efficiency.
9. What is the basic principle of design for an oil burner?
10. Name four types of oil burner.
11. If you were designing a plant for a ship, what kind of burner would you use?
12. What kind of burner would you prefer for your own house? Why?
13. What is a sprayer plate? With what type of burner does it function?
14. How do you size an oil burner?
15. Describe the ignition and safety system of a gas-electric firing sequence.
16. How does a vaporizing burner differ from other oil burners?
17. What is an electric eye? How does it function?
18. How does a gas burner work?

CHAPTER 7

WATER HEATERS AND HOW TO SIZE THEM

Direct-fired Heaters

Bucket-a-day Stoves. In the midst of a quiet afternoon, the roof of a high-school boilerhouse suddenly lifted several feet into the air, window glass burst from the frames, and the entire community shuddered with apprehension as bits of iron were flung almost as far as the sound.

The explosion originated in a familiar type of water heater known as a "bucket-a-day stove," the bucket being a bucket of coal. It seems that the janitor fired the stove but failed to open a circulating water valve in the pipe line between the stove and the storage tank, thus turning his stove into a generator of high-pressure steam. The janitor was charged with carelessness, but the real culprit was the engineer or contractor who had installed such a system without a safety valve on the heater.

Every water heater should be served by a safety valve, and all of us who design, install, or service hot water systems should make this a cardinal rule for our guidance. A domestic water heater of any type has explosion possibilities, and the fact that many cities do not insist on safety valves for domestic water heaters does not excuse us for being guilty of an unsafe practice.

Recent studies of water-tank explosions indicate that disaster tends to follow excessive temperatures rather than excessive pressures. For example, a tank being tested at 150 lb pressure and 195 F failed to explode even when prodded with a sharp instrument that eventually pierced its side. The perforation released a squirting stream of hot water, and nothing else happened. In contrast, a similar tank containing water at 265 F exploded with great violence although the pressure was only 50 lb at the time. In this latter test of course, 265 F meant that potentially the water was now steam, and the tank was charged with an expanding energy far beyond its power to restrain.

These phenomena have been duplicated often enough to substantiate the warning that whenever water reaches the boiling temperature regardless of pressure it becomes deadly dangerous and some form of emergency relief is essential.

Gas-fired Instantaneous Heaters. Figure 7.1 illustrates a heater burning gas. In this unit we observe the operation of what is termed "instantaneous heating." There is no storage tank; the copper coil receives cold water at one end and discharges hot water at the other. The heat is applied to the coil only during those periods in which a hot water faucet somewhere in the house is open. If we shut off the faucet, the main gas flame is extinguished simultaneously, leaving the pilot to stand by in readiness for the next demand. The operation of this heater is controlled by a thermostatic element in the heater, and is readily adjusted to deliver water of any temperature that we want.

Instantaneous water heaters of this kind are quite satisfactory, although the little pop that accompanies the pilot ignition of the main burner may be disconcerting every time we turn on a faucet, until we get used to it.

Gas-fired Storage Heaters. Figure 7.2 shows a gas-fired storage type of heater in which fire flues are submerged in the tank. The hot gases of combustion from the burning fuel pass through these submerged tubes and the surrounding water is heated thereby.

An interesting approach to the gas-fired heater design is illustrated by Fig. 7.3. According to the builder, a reduced heat loss is achieved by shaping the tank like a ball, because the external surface of this ball exposes 18 per cent less area than does a conventional cylindrical tank of equivalent volume.

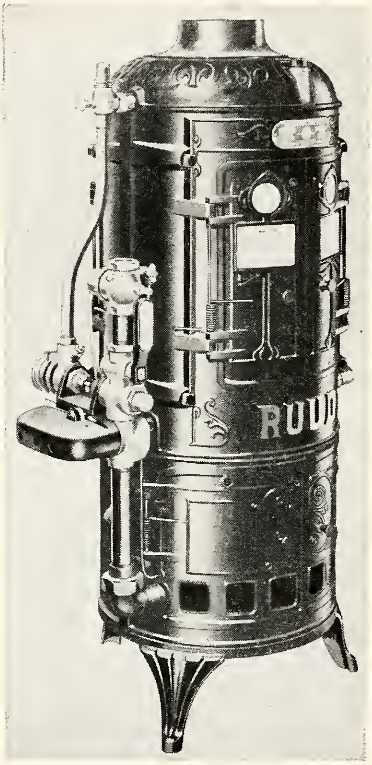


Fig. 7.1. Instantaneous type of gas-fired water heater. (*Courtesy of Ruud Manufacturing Company.*)

The internal arrangements of most heaters vary with the ideas of their builders, all aiming to produce plentiful hot water with a minimum of fuel. From time to time equipment will be observed that is fitted with ribbon-like helical strips of metal inside the fire tubes, called "retarders" or "gas twisters" or some other appropriate name. The purpose of these ribbons is to give the hot gases of combustion a spiraling or rotating motion and thereby increase the convective heat exchange through the surfaces over which they pass. This design appears to have a couple of disadvantages, for example, it increases the draft loss through the heater, and again sometimes acts as a notable soot catcher if combustion is bad. However, nothing is perfect, as most of us know; and if retarders are economically profitable, then we are for them.

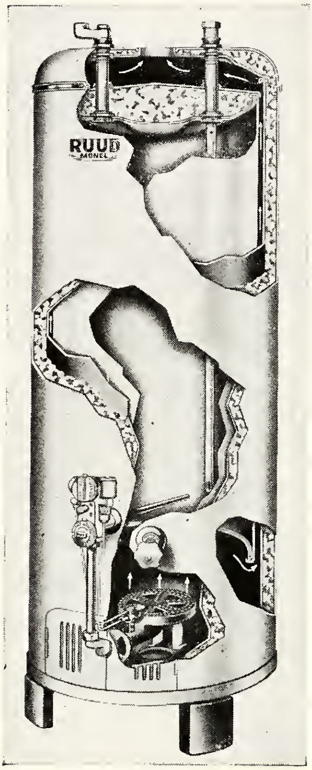


Fig. 7.2. Gas-fired storage heater. (*Courtesy of Ruud Mfg. Co.*)

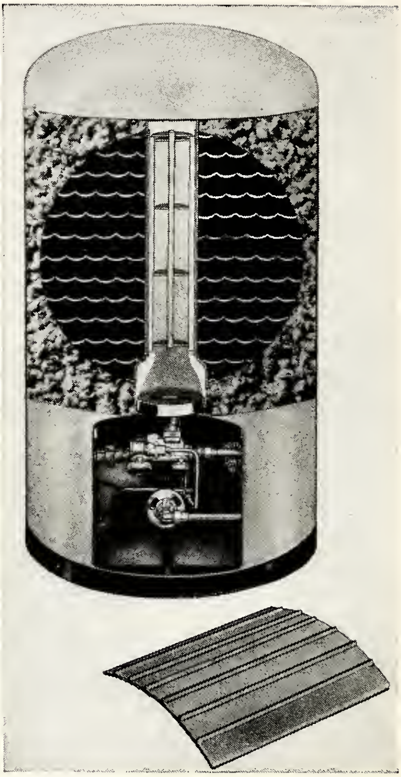


Fig. 7.3. Spherical or ball-type storage heater, gas-fired. (*Copyright by Servel, Inc.*)

Oil-fired Storage Heaters. The heater internally fired by oil is illustrated by Fig. 7.4. Oil-fired heaters may be ignited from an oil pilot, gas pilot, or electrically.

The burners for the oil-fired heaters include gun types, rotary types, and the simple pot design in which oil trickles into a pot or dish, being admitted by a flow control valve outside the pot. Air to the pot burner frequently is supplied through small holes in the wall of the pot, under pressure from a supplementary blower.

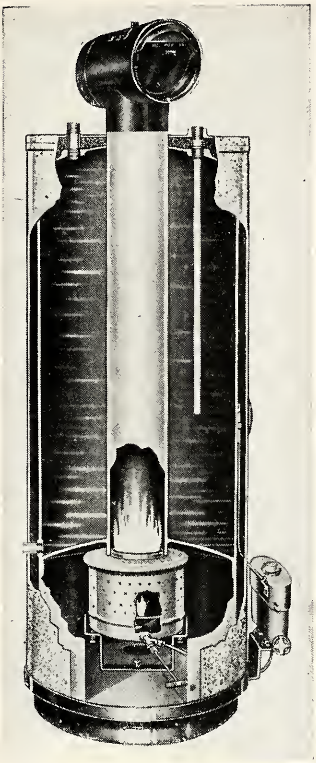


Fig. 7.4. Storage heater, oil-fired. (*Courtesy of Ex Cel Company.*)

Fig. 7.5a. Range boiler as manufactured by Scaife Company.

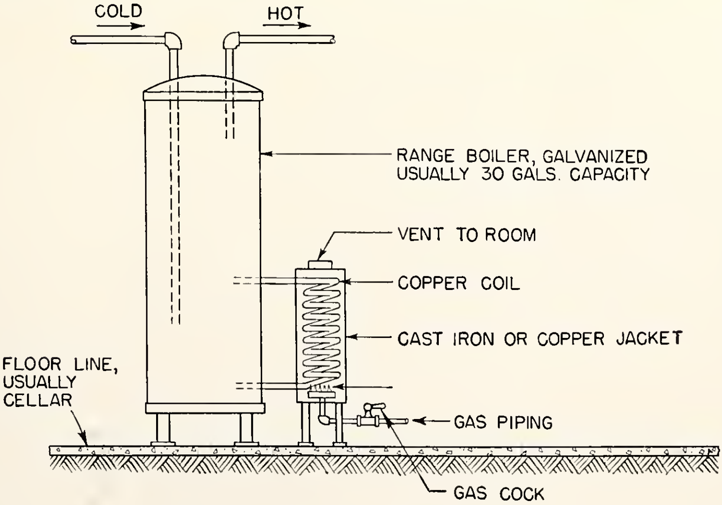

COLD
HOT
RANGE BOILER, GALVANIZED
USUALLY 30 GALS. CAPACITY
VENT TO ROOM
COPPER COIL
CAST IRON OR COPPER JACKET
FLOOR LINE, USUALLY CELLAR
GAS PIPING
GAS COCK

Fig. 7.5. Sidearm heater with range boiler.

As all these burners are described and pictured in Chap. 6, we will not expand on their functions here.

Nonintegral Direct-fired Heaters. The bucket-of-coal-a-day type of stove, which produced the violent explosion we discussed earlier in this chapter, is an example of the nonintegral direct-fired water heater. So is the long remembered "range boiler" illustrated by Fig. 7.5, in which the fuel happens to be gas. Many of us now own, or will recall having had, such a water heating system in our houses. When we wanted to take a bath, we descended to the basement, or perhaps went into the kitchen wherever the little giant happened to be, lighted a match, turned on the gas, and sat down to read a book while the vertical galvanized tank beside the heater was being stocked with hot water.

Some of us may regretfully remember having forgotten to go back and turn off the heater. In fact, I once observed a forgetful gentleman sitting on exhibition in his bathtub, the front wall of the bathroom having been carried away and into the public street by the exploded and soaring tank.

Nonintegral direct-fired heaters are no more dangerous than any other heater. Any heater can explode if the proper safety devices are not present.

Nonfired Heaters

The heaters which generate hot water without application of flames are termed nonfired, or more frequently, indirect heaters. In this group we find the following designs:

1. Built-in, tankless heaters, mounted inside the house heating boiler.
2. Instantaneous, external tankless heaters, with steam as the heating agent.
3. Solar water heaters.
4. Electric water heaters.
5. Any of the above, supplemented by a storage tank.
6. Open heaters, in which steam is admitted to a tank or tub of water.

Tankless Heaters, Built-in. The built-in tankless heater is illustrated by Fig. 4.4. It takes the form of a coil, or loop of pipe, submerged in the water of the boiler. Hot water from a built-in heater tends to approximate the boiler water in temperature, and if we find this too hot, a mixing valve should be installed which will introduce cold water into the hot water piping and thus reduce the delivery temperature. Mixing valves operate automatically on the thermostatic principle, so that once installed, and set for the temperature we want, there is nothing more for us to do.

Built-in heaters have the disadvantage that, during summer months, we must fire the house heating boiler just to obtain hot water. In these cases, a substitute heater of some kind is quite desirable, and frequently installed.

Tankless Heaters, External. The instantaneous tankless heaters are illustrated by Fig. 7.6. Inside the shell of this heater is a coil which is kept hot by a surrounding blanket of steam or hot water originating in the main boiler. Water to be heated passes inside the coil, absorbs heat from the hot blanket and, if a faucet is open, emerges warmly at that point.

These external tankless heaters and the piping which connects them to their boilers should be insulated to conserve heat. The shell surface is nearly as hot as the steam or water within, and this loss of Btu to the boiler room is of no value to anybody.

Solar Water Heaters. Solar water heaters are commonly used in those parts of the country where the sunlight is plentiful and hot. Mostly they are mounted on the roof, occasionally on the ground, and are tilted to face the sun over the greatest possible number of hours. A solar water heater is illustrated by Fig. 7.7, which shows the arrangement of the component parts.

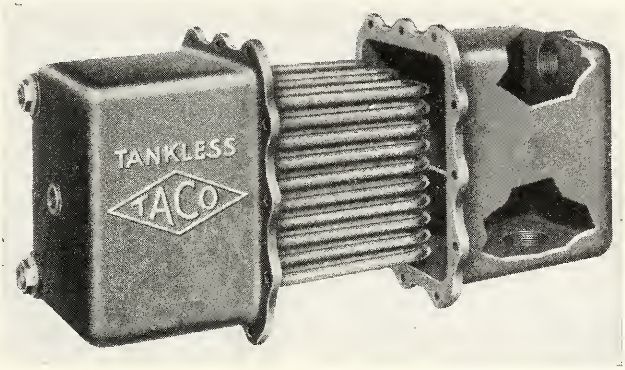


Fig. 7.6. External tankless heater. (*Courtesy of Taco Heaters, Inc.*)

The idea of capturing heat from the sun is most attractive, but practical development of suitable equipment is not altogether simple. The tilt of the box, for instance, has considerable influence on the efficiency of the arrangement, and 10 deg from the horizontal may be just right in the summer, but all wrong in winter. In the latter season, 30 to 45 deg might be far more efficient. In addition to this seasonal variation in tilt, we have the variation resulting from latitude, so that as we approach the equator, the tilt tends more and more to become horizontal. The purpose of the tilt of course is to keep the direct rays of the sun on our absorbing surface as long as possible.

The absorbing unit consists primarily of a copper coil soldered flat to a copper plate, and set inside of an insulated box, with everything under the glass cover painted dull black. The black finish to the copper plate, tubes, and inside surfaces of the box takes advantage of the fact that black is the best absorber of heat we know, and the ability of any other color to perform this service is measured against black as a standard.

The bottom and sides of the absorbing box are insulated with an inch or two of cork, mineral wool, sawdust, or other suitable material to check loss of the captured heat. The glass cover sometimes is made of single thickness window glass; sometimes it is formed of double panes with an air space between them. This air space performs an insulating service similar to that of the materials elsewhere insulating the box, and if we wonder why an air space can prevent heat from getting out while permitting its entry, we find the answer in our examination of solar energy. Solar energy is radiant energy, and heat as we know it is generated when this energy collides with a surface which absorbs it. Thus the glass cover, permitting the passage of the sun's rays, witnesses the conversion of these rays into heat as they strike against and are absorbed by the black surfaces of the plate and piping. Now the problem is to keep this newly generated heat inside the box, and the double glass cover with its air space between is comparable to the storm sash which we set up to check heat loss from a window in winter.

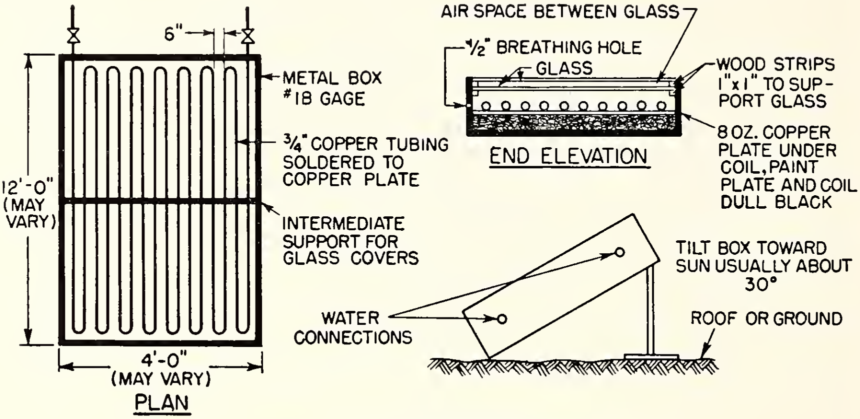

6"
METAL BOX #18 GAGE
3/4" COPPER TUBING SOLDERED TO COPPER PLATE
12'-0" (MAY VARY)
INTERMEDIATE SUPPORT FOR GLASS COVERS
4'-0" (MAY VARY)
PLAN
AIR SPACE BETWEEN GLASS
1/2" BREATHING HOLE
GLASS
WOOD STRIPS 1"x1" TO SUPPORT GLASS
8 OZ. COPPER PLATE UNDER COIL, PAINT PLATE AND COIL DULL BLACK
END ELEVATION
TILT BOX TOWARD SUN USUALLY ABOUT 30°
WATER CONNECTIONS
ROOF OR GROUND

Fig. 7.7. The solar water heater.

All solar heaters, however, are not built with double glass covers. This kind of construction adds to the cost, and many owners are satisfied with single glass efficiency as the whole unit is a fairly expensive contraption anyway. The economy, of course, results from getting hot water free once the system is installed. Thus over a period of years, a solar water heater can well be the cheapest to own, despite the high initial cost.

Solar heaters require storage tanks because an accumulation of hot water for nights and cloudy days is essential. The tank must be located well above the absorbing box, with the bottom of the tank connected by piping to the bottom of the coil in the absorber. The hot water leaving from the top of the absorbing coil is then piped to the top of the storage tank. This kind of circuit produces the well-known thermosiphon type of circulation, which results from the greater density of the cold water in the bottom of the tank pushing its way down and into the absorber coil, and thereby forcing the lighter hot water out of the coil and up to the top of the tank. A high static head augments this circulation most desirably.

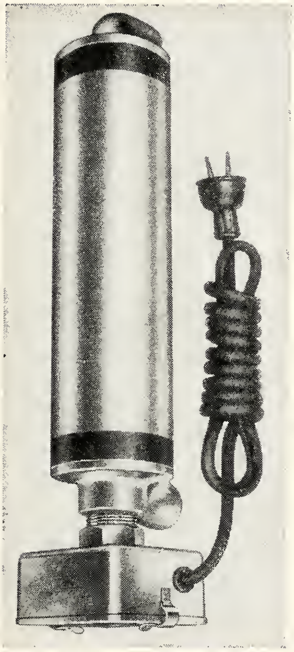


Fig. 7.8. Electric sidearm heater. (*Courtesy of the Cady Co.*)

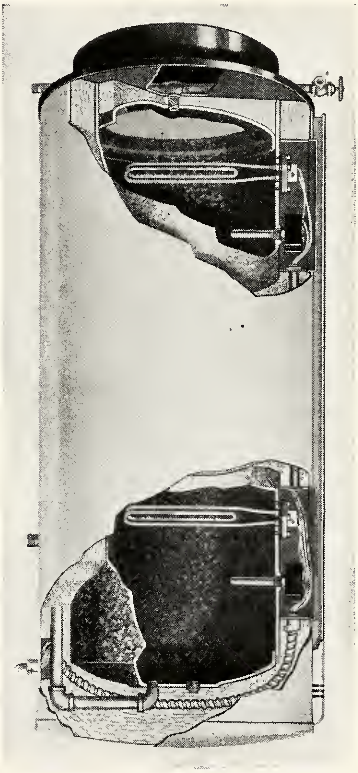


Fig. 7.9. Storage heater using electric immersion elements. (*Courtsey of Ex Cell Company.*)

One more note about solar heaters and then we shall leave them. This note concerns their size. Each square foot of the glass cover will pass enough solar energy to produce, on the average, from 1 to 1.8 gal of hot water per day, the variation being due to the seasons, the tilt of the box, and the intensity of the sunlight. The water output will approach 140 F in temperature, although under extra favorable conditions much hotter water frequently is delivered. A commonly observed size is 4 ft wide by 12 ft long, using $\frac{3}{4}$- or 1-in. copper coil. The initial cost of such a system might be somewhere around $200, including storage tank and piping.

Electric Water Heaters. Electric water heaters normally fall into two groups: (1) the heaters with the heating elements immersed in the tank and (2) those with the heating element off to one side of the tank. These latter resemble the range boiler heater in the matter of piping connections to the storage tank, as a continuous thermosiphon circulation wends its way from the bottom of the tank to the heater and then back to the top of the tank. Figure 7.8 illustrates this sort of heater, and incidentally, such heaters, whether using electricity, gas, or oil, are sometimes called "sidearm" heaters. The term is quite appropriate because certainly the heating unit is off to the side and the thermosiphon circulation makes it an arm of the storage tank.

The immersion heaters, as shown in Fig. 7.9, usually have two sets of heating elements, one in the top of the heater and one in the bottom. As the two sets never are on the line simultaneously, good economy and reduced wiring sizes are possible.

Table 7.1 shows the current requirements for the various groups of elements in heaters of different capacities, together with wire and fuse sizes.

Temperature controls are integral parts of electric water heaters, and we have no difficulty in obtaining the degree of heat we want. We set the control to suit ourselves.

TABLE 7.1. SIZES OF WIRING AND FUSES FOR ELECTRIC HEATERS

Heater size, gal	Upper element, watts	Lower element, watts	Wire size, heater circuit	Size of fuse, amp
40	1,250	750	14	15
50	1,500	1,000	14	15
60	2,000	1,250	14	15
80	2,500	1,500	14	15
110	3,000	2,000	12	20
120	4,000	2,500	12	20
140	4,000	3,000	12	20

Note: Local building or electrical codes should be consulted in all cases.

Auxiliary Furnace Coils. The idea of getting something for nothing frequently actuates those of us who stick a water pipe, or coil, in the back of a heating furnace which never was designed for such a contraption. We reason that the heat is going up the chimney anyway, so why not capture some of it for use in our showers and tubs?

Unfortunately, fuel is not free, and although perhaps we get our hot water all right, the price is paid, like hidden taxes, in the extra coal, gas, or oil which we burn without generally realizing that we are burning it. This happens in the majority of cases, where the heater has been sized to heat the house, and nothing more.

If we have an oversized heater and are burning fuel wastefully because of it, then a water coil might be of value. The best way to tell is to check both the chimney-gas temperature and the concurrent percentage of CO_2. If we find the combustion efficient and the stack temperature way up, say higher than 600 F, then a carefully sized furnace coil might pay off. On the other hand, introducing additional cold water surface into the furnace may so reduce the furnace temperature that combustion is retarded, and both smoking and a drop in efficiency will result.

In short, if we want a coil in the furnace, the best way to get it is to buy a heater properly sized for the house and which, in addition, contains a coil correctly designed and located in the boiler by the manufacturer.

Open-pipe Heaters. Occasionally, in process and restaurant activities, we will find a small steam pipe, usually about ½ in. in diameter, carrying steam to an open basin or tub. The pipe may have simply an open end, from which the steam emerges to mix with and heat the water. More elaborate designs may present a perforated pipe submerged in the water either horizontally or vertically, and consequently a much more uniform heating is obtained as a result of the escaping steam being distributed throughout the body of the water.

The steam required for such water heating is computed quite simply. For instance, suppose that we have a tank containing 500 gal of water at, say, 50 F and we want to heat this water to 180 F. For the heating medium we have saturated steam at 2 psig.

Now 500 gal of water at 50 F weighs about 8.3 lb to the gallon, giving us a total water weight of 4,150 lb. Since 1 Btu will raise 1 pound of water 1 degree Fahrenheit, $4,150 \times (180 - 50) = 539,500$ Btu, which must be supplied by the steam.

The steam tables show the total heat of saturated steam at 2 psig to be approximately 1152 Btu, and the heat in water at 180 F to be 148 Btu. The heat difference between the steam and the water, amounting to 1004 Btu, will go into our tank of water, and $539,500 \div 1,004$ indicates we shall require 537.4 lb of steam to accomplish this heating chore.

How to Choose a Water Heater

Basic Considerations. In most cases, the choice of a water heater is based on these considerations:

1. Fuel available.
2. Cost of fuel available.
3. Initial cost of equipment.

Suppose, for example, that we live in the Pacific Northwest. Here electricity is comparatively so cheap that it is being used industrially for the generation of high-pressure steam. At Salem, Ore., an electric boiler produces 150,000 lb of steam per hour at 150 psig, and the equivalent total electric rating of this steam generator is 50,000 kw.

With evidence so weighty as this before our eyes, we certainly will consider electric water heaters most seriously before making a decision.

Now suppose that we move to Florida, where the sun shines long and hot. Looking around at our neighbors' houses, we will observe the glass-covered boxes of solar heaters on many roofs. As these units require no fuel at all, we certainly must consider solar heating with all care.

In other parts of the nation, natural gas, manufactured gas, liquefied petroleum gas (generally called "bottled gas"), coal and oil, all present their peculiar local advantages. What we finally choose then is likely to be the common custom of the locality.

The economic picture, based on an equipment life of 10 years (the period most commonly used) is shown in Table 7.2.

TABLE 7.2. METHOD BY WHICH THE 10-YEAR OWNING COST OF ANY WATER HEATER MAY BE APPROXIMATED*

Type of heater	Initial cost	Fuel cost for 10 years	Maintenance cost for 10 years	Investment interest, 2%	Total cost for 10 years
Electric	$100	$1,000	$50	$20	$1,170
Gas fired	100	750	50	20	920
Oil fired	100	600	50	20	770
Coal fired	100	600	50	20	770
Tankless:†					
In boiler	50	500	25	10	585
External to boiler	85	500	25	17	627
Solar	200	......	25	40	265

* *Note:* This tabulation does not represent comparative costs of equipment or of fuels, and must not be interpreted that way. Its sole purpose is to show the method by which the 10-year-owning cost of any water heater may be approximated. In each case, local prices for equipment and fuels should be used.

† Heating season only. Auxiliary heating charges must be added for summer supply of hot water.

For estimating the fuel cost in the foregoing tabulation, we must of course make several approximations, and must consider weightily the characteristics of the hot water patrons. We can do this by following the directions now presented on sizing water heaters.

How to Size Water Heaters

Sizing Domestic Heaters. The first rule for sizing a domestic hot water installation concerns the number of users to be supplied. The importance of this rule is realized hundreds of times by persons who finish a thorough soaping, and then discover there is no hot water for rinsing.

Those of us responsible for these sticky predicaments generally can repeat the rule in our sleep ever after: Never size a water system for 30 persons if 31 will use it.

The second rule is likely to be realized most poignantly by those of us who are trying to alibi our breaking of the first. In our own rueful words, it can go as follows: Never accept as sizing law the published tables on hot water requirements per faucet, per bathroom, per kind of building, or per anything else, but determine every job on its own merits.

Residential Considerations. What are the merits of an individual job? Suppose that we consider residences first. We must evaluate the following conditions:

1. How many persons will be using hot water? Not how many bathrooms are in the house, or how many bedrooms, but how many persons. The importance of people is illustrated by a certain three-story residence that was converted into an institutional girls' dormitory, with up to four girls in one room, and all using one bathroom, one tub. Obviously, hot-water demands estimated on anything but persons would be disastrously wrong in this case, and the fact that such overloading of a bathroom is unsanitary and perhaps illegal is not an acceptable alibi for a miscalculation.

2. Is there a home laundry? Generally, the laundry peak will not coincide with the peak for everyday living, but we must be sure that our tank will satisfy the laundry demands, particularly if one of the new automatic laundries adorn the premises. These new units tend to use more hot water than the old-style washers.

3. What sort of persons make up the household? A child five years old or younger is the equivalent of two adults when we are figuring demands for hot water. And how about domestic servants? The first servant in a home equals two adults; additional servants are added to the load as one adult each. Does any member of the family work in a shop, or elsewhere regularly acquire a kind of grime that is hard to remove? If so, wisdom suggests that we figure him as two adults. The determination of the equivalent number of persons is summarized in Table 7.3.

4. How much entertaining does this household do? If we do not already know it, our wives will inform us that overnight guests go right into our soap, hot water, and linens. If they dine with us too, the result-

ing mountain of dirty dishes in the kitchen sink will keep the hot water generator in action long after the normal shutdown time. We must allow something for entertaining by people who live in big houses, and as these are in the minority, we class them as group B. The two- and three-bedroom house we shall call group A, as this type houses the bulk of our population. Group A homes conserve on entertaining as well as on materials, dinner for ten being quite a problem in a dinette, and sleeping on a sofa being unattractive to most hosts.

5. How about the future? We are not always able to predict the arrival of babies or of in-laws, but it is wise to keep these possibilities in mind. In fact, some appliance dealers think a water heater should be sized for probabilities of increase during the next five or ten years. Here we must exercise our judgment. If a family with one child tells us their goal is four or six, a discount of 50 per cent might be allowed, but no more. Some parents really mean it.

TABLE 7.3. EQUIVALENT NUMBER OF PERSONS USING HOT WATER*

Type of Individual	*To Be Counted as Persons Using Hot Water*
Adult	1
Child, under six	2
Child, six or over	1
First domestic employee	2
Each additional domestic	1
Mechanic, greasy hands	2

* *Note:* These figures are to be used for residential calculations in conjunction with Table 7.4.

Table 7.3 summarizes the situation for residential estimating, with Table 7.4 showing the minimum heater sizes recommended.

Sizing for a Factory. Sizing hot water generators for commerce and industry introduces new factors and complicates some of the old. Here are the conditions we must evaluate:

1. How many persons will wash and what kind of work do they do? Suppose that we have a factory with 20 persons in the office and 80 in the shop. According to Table 7.5, which has been compiled from several sources, we might expect each lavatory basin to draw 3 gal per hr. However, we know that the shop personnel will require more water than the office personnel, individually. When the noon whistle blows, we can expect each of the shop men to draw 2 qt of hot water for his hands, 2 qt for his face, and 2 qt for rinsing, a total of 6 qt, or 1½ gal. Multiply this by 80, and the answer is 120 gal. Meanwhile the office folks will wash their hands thoroughly, but less than half of them will do any more than casually dab at their faces. Three quarts will be a reasonable average

Table 7.4. Minimum Safe Sizes of Water Heaters, for Residential Service, Gallons Capacity*

Equivalent number of persons	Group A family	Group B family
3 or less	40	50
4	50	60
5	50	60
6	60	80
7	60	80
8	80	50 (2)
9	80	50 (2)
10	50 (2)	60 (2)
11	50 (2)	60 (2)
12	50 (2)	60 (2)

* *Note:* These sizes are recommended for all normal household installation, although compiled originally from NEMA standard heaters. Judgment must be exercised in allowing for future increases in the demand.

Table 7.5. Peak Hourly Demands for Hot Water
(Data are in Gallons of Hot Water per Hour per Tub, per Basin, per Shower, etc.)

Building	Tubs	Showers	Private basins	Kitchen sinks	Laundry tubs	Slop sinks	Public basins
Apartments	15	60	3	10	25	20	4
Clubs	18	200	3	20	30	20	6
Dormitories	30	200	3	. .	35	15	6
Gymnasiums	30	200	2	. .	. .	. .	8
Hospitals	15	60	4	20	35	20	6
Hotels	20	75	3	20	35	30	8
Industrial plants	30	200	6*	†	. .	20	10
Office buildings	. .	. . .	3	†	. .	15	6
Homes	15	60	2	10	25	15	
Schools	. .	200	3	†	. .	20	15
YMCA	30	200	3	†	30	20	10
YWCA	30	150	3	†	30	20	10
Auto courts	20	75	3	. .	35	. .	8
Retail stores	. .	. . .	3	. .	. .	30	10

* The basins considered here are those in the shop washrooms which are used by shop personnel. For office workers in an industrial plant, use the office building figures.

† Refer to Table 7.7, which provides data for restaurants and cafeterias.

for each of them at noon. Some, carrying their lunches, will not wash until later, thereby eliminating themselves from the peak hot water load. Therefore 20 persons at an average of 0.75 gal each will contribute 15 gal to our noon peak. This gives us a total of 135 gal for the noon washing load. Now if we simply had estimated 3 gal for each of the, say, 25 hot water faucets, we would be equipped for 75 gal only.

2. Is there a restaurant or cafeteria in the factory? Estimating hot water loads for a restaurant is rather complex business, so much so in fact, that we shall examine the process in detail later on. Meanwhile, for the purpose of determining a factory load, suppose that we assume something for the factory cafeteria, say 180 gal, and then proceed to the other elements.

3. What will they need for the plant hospital, or first-aid room? Sterilization of instruments is accomplished electrically for most first-aid rooms nowadays; therefore we need figure only on miscellaneous washing. We must talk with the doctor, or nurse, or other person in charge, to evaluate this load. Generally, we can expect to absorb the medical and accident demands in our equipment's peak capacity, but if there is any possibility of a sizable demand, we had best allow something for it. Soap left on the skin by a rinseless shower is annoying; but grease and grime left for long on an amputated or crushed finger may create a job for the undertaker.

4. What are the process demands? Many plants will have no process uses for hot water; others require it irregularly or at long intervals. For instance, during the past few years, a procedure for washing the soot and scale from the outside of boiler tubes has been developed with considerable success. Hot water, say 200 F, and at a pressure of 150 psig is directed against the tubes, after the boiler has cooled. This practice uses a great deal of hot water, but so infrequently during a year's time that a steam and water mixing nozzle generally will be preferable to an oversized generator. Whatever the process load happens to be, we must evaluate it carefully from the equipment sizing viewpoint. We must also consider the time of peak demand. Frequently, process demands will shut down during the noon hour and then will pick up for a short time and coincide with the restaurant dishwashing load. However, as the restaurant load will be falling off, a storage tank may provide the supply for these brief peaks, another item to consider.

Estimating the Peak Hour. A summary of the noon load of the factory we have used as a "guinea pig" is given in Table 7.6.

Now we must consider a second peak, the peak that occurs at the end of the working day. Most of our office personnel will leave without washing, eager to clean up and change at home. We can safely assume the five o'clock peak at one-half the noon peak for these people. Out in the shop, the situation will be quite different. There we can expect to find the shower baths in full operation. With 80 men on the payroll, we may find 4 showers in use. Table 7.5 indicates 200 gal per hr will be squirted from each fixture. Although it is possible that a shop man would wait an hour for his turn in a shower, it is hardly probable. We can expect the showers to shut down for the night 20 min after they start. This means 67 gal will be used by each shower, a total of 268 gal.

TABLE 7.6. SUMMARY OF NOON WATER LOAD

Noon Load	
Total washing	135 gal
Restaurant, estimate 1/5 of max	36 "
Medical	...
Process	...
Total noon load	171 gal
After-lunch Load	
Washing, estimated for hand rinsing (office personnel only) 10% approx. of noon office peak	2 gal
Restaurant, peak dishwashing load	180
Medical	...
Process	...
Total post-lunch peak	182 gal

Perhaps 20 of the men will use the showers; the others will wash their hands and faces, generally somewhat less thoroughly than at noon, because shop people like to get home too. If we provide 1 gal for each of these hands and face washers, we need a total of 60 gal more of hot water. Add this to the shower and office load, and our five o'clock peak turns out to be the high point of the day, with 336 gal bursting from various kinds of fixtures.

Now this is the way the situation sizes up theoretically, and if we must, we can proceed accordingly. However, there is rarely a problem which is not solved elsewhere, and before making a final decision, common precaution suggests that we visit the other fellow's factory and observe for ourselves the washing habits of the people who run it. For example, if our proposed plant is going to make ball bearings, the ball-bearing plant at the other end of town, or in the next county, will have the same kind of dirt and grease to remove, and there will be no difficulty in obtaining from the operating plant data on the number of persons, the size of their hot water generators, the number of showers, and if there is ever anybody who ends up with nothing available but cold water.

This kind of information can be made to help tremendously, for with it in hand, to check against our theoretical figures, we can say, "We *know* how much hot water we will need."

Water Heaters for Restaurants. Estimating hot water for restaurants requires us to evaluate a series of variables. There are approaches to the problem galore, including the restaurant cubage system, which advocates basing hot water figures on the cubic feet of space, counting the kitchen too. That this system has some value becomes less surprising when we remember there is a direct relationship between space and the number of customers who can be handled within it.

Another approach, advocated as quick and easy by its proponents, merely directs us to allow some empirical number of gallons per hundred customers—but there is a difference of opinion on the empirical number. One authority mentions 300 gal for 500 persons; another expands the relationship to 220 gal for 200 persons.

The ASHVE Guide suggests the value of the average check as an indicator of the dishwashing required, and there is a great deal to be said for this approach. In counterrooms and diners, where the average check is low, the dishes and silver in use will satisfy the essentials and little more; nobody puts out butter knives, for example, in a hamburger stand, even when it is spelled shoppe. At the other end of the scale, hotel dining rooms serving patrons at $3 up, and night clubs where prices are topped only by the sky, will serve meals that start with cocktails and end up with finger bowls for those able to survive what they have had in between. Obviously, as the price increases, so does the dishwashing.

Table 7.7 is a carefully weighted table, based primarily on the number of patrons, and modified to compensate for the varieties of luxury experienced in a variety of restaurants. Much of the data provided in Table 7.7 is the result of many years of personal investigation into the operations of all classes of restaurants. Restaurants were asked how much water was used, how many persons were served in a peak hour, and how much was the average check.

In using Table 7.7, some allowance may be necessary at times for local water conditions. If city water is hard, and softeners are not contemplated, more water will be needed for adequate washing and rinsing.

Hot Water for Office Buildings and Hotels. In estimating hot water needs for office buildings and hotels, we have, as usual, several choices. We can itemize every faucet and every other type of fixture and add up their individual demands from Table 7.5, and then apply a diversity factor. Alternatively, we can base our calculations on the number of occupants.

Notably, we are quite sure to reach nothing but trouble if we attempt to base our estimates on cubage. There is no uniformity in the way cubage is devoted to persons in these structures. The National District Steam Association has made exhaustive studies in a dozen cities on the use of steam for generating hot water, and has then attempted to relate their findings to cubical contents. The average for office buildings is reported at 0.7 lb of steam per cubic foot of building; but the range flutters from less than 0.3 all the way up to about 1.80, with plenty of representatives at all the various levels in between. Obviously, on the cubage basis, we might as well close our eyes, and choose the first number that occurs to our tired brains.

TABLE 7.7. HOURLY HOT WATER USAGE IN RESTAURANTS*

Group symbol	Type of restaurant	Usual type of washing	Avg, gal/meal
A	Units in large chains, serving from 200 to 300 or more persons per hour, mostly at tables. Checks range from 65 cents at lunch to $2 at dinner	Machine	1.94
B	Lunch counters in department stores and five- and ten-cent stores, serving limited platters	Machine	1.50
C	Soda fountains in drugstores and candy stores, serving sandwiches and coffee	Hand	1.00
D	Drugstores with fountains and considerable table service, serving dinner platters up to about $1.25 each	Machine	1.30
C	Grilles with steady flow of bar drinks and a lighter food sale	Hand	1.00
C	Sandwich stands, diners, etc	Hand	1.00
D	Cafeterias, large, with wide choice of hot dishes	Machine	1.30
C	Cafeterias, small, factory cafeterias	Hand	1.00
A	Hotel coffee shops, serving from 75 to 150 persons per hour, with dinner checks up to $2 each	Machine	1.94
E	Hotel dining rooms, serving full course dinners with finger bowls, up to $4	Machine	4.00
F	Commercial hotel dining rooms, average dinner-check range from $1.50 to $2.75	Machine	2.50

* *Note:* Generally, the amount of hot water used increases with the increasing scale of dinner prices because more plates are used and more care is taken to wash them efficiently in the higher priced restaurants.

These unit figures have been compiled from several sources, and represent averages of observed restaurants. Range variations in each group may run from 25% more than the average to 10% less than the average.

Estimates based on the number of persons are, perhaps, the most dependable in large buildings, as well as elsewhere. For instance, 10 faucets being used by 100 persons would certainly run short if we allow, say, 3 gal per faucet; conversely a situation of 100 faucets and only 10 persons could mislead us into providing a deluge when all we need is a drizzle.

For hotels and large apartment houses, the ASHVE Guide suggests the assignment of 40 gal per day, per person. Other sources of experience mention 30 gal.

For office buildings, banks, and similar commercial structures, the range per person recommended by various authorities is from 2 to 3 gal per day.

While these variations may appear a little puzzling, actually they are all correct, for each reported conditions reflects the findings of the investigators modified, perhaps, by factors of judgment. The area of the observations and the time of the year definitely affect the findings; an industrial city, for example, where the dust-soot fall in winter approaches say 80 tons per month per square mile, certainly may be expected to show a higher water usage than a tourist town in Florida. As usual, we shall need to exercise our judgment, and when in doubt, allow a little more, rather than a little less.

The question of how much storage capacity to provide is always a sticky question. How we determine the answer depends on the following factors:

1. How will the hot water be generated?
2. Is the load a steady one, or intermittent?

Let us consider factor 1 first. If we install a direct-fired water heater, and the load is rather steady, then we shall probably have sufficient reserve capacity in the heater to meet small, infrequent peaks, and need figure on no other storage. On the other hand, with sharp peaks, and a generally up-and-down demand, economy of operation will indicate the need for a storage tank which, added to the heater output, will meet these peaks.

Hot Water from District Steam

Demand Charge. If we buy steam from a district steam system for making hot water, then we must consider the factor of maximum demand charge. For instance, if steam demand is billed at say $1 per pound of demand per year, then a demand of 1,000 lb, which would heat approximately 1,200 gal of water 100 deg in 1 hr, would cost us $1,000 per year. Incidentally, 1 lb of steam will heat about 10 lb of water, roughly 1.2 gal, if we assume that a gallon of hot water weighs 8.2 lb.

Obviously, to keep down the demand charge, we should spread the generation of 1,200 gal over as many hours as we can. If the consumer is an office building, we can expect at least 3 hr of negligible draw between peaks, perhaps longer if there are no especial requirements in the building, such as beauty parlors, barber shops, etc. Thus we have 3 hr for accumulating hot water in a storage tank before each peak.

Apparently, 300 gal per hr from our heater should do; however, we can count only 75 per cent of the storage capacity of a tank, owing to the progressive dilution by incoming cold water. Consequently, we must size our heater as follows:

Input to tank in 4 hr	1,200 gal
Available on demand (75%)	900
To be made up by heater in peak hour	300 gal
For total 4-hr load, add tank input	1,200
Total 4-hr load on heater	1,500 gal
Average hourly load on heater	375

Since 1.2 gal of water will be heated by 1 lb of steam, 375/1.2 = 313 lb of steam, approximately, will be drawn from the district steam mains, and our demand charge drops from $1,000 a year to $313. In these circumstances, a storage tank will pay for itself in a single year, or very soon thereafter.

If our hot water is to come from the building boiler plant, then we must be sure that the capacity of the boiler or boilers is sufficient to satisfy this extra load. We can approach the problem as we did in the case of the district steam. If the same load is used, for example, 313 lb of steam will be drawn from the boiler each hour. For those of us who think in terms of EDR, this means an extra load on the boiler of approximately 1,252 sq ft; for minds that deal in boiler horsepower, about 10 will do it.

Coming back to factor 2, which concerns the steady or intermittent nature of the load, we observe that the decision to store or not to store will rest mainly on this factor. A uniform load offers no striking advantage through storage; if we must have 1,200 gal an hour every hour, we might as well generate it as we use it, and thus save the cost of a tank.

The relationship between total daily use and the maximum hour is another of those variables which involve our judgment. The ASHVE Guide estimates that in the case of apartments, hotels, and residences the maximum hourly demand will approximate one-seventh of the total daily use. The same authority suggests one-fifth for office buildings and one-third for factories.

These figures are probably the most dependable available for general estimating, but in each case they must be weighed with all care. The larger the building, whether hotel or office, the greater is the tendency for the peaks to level out. In skyscraper office buildings, for example, the maximum hour relationship may decrease to one-eighth or one-ninth of the daily load, and approach the relationship observed in hotels. This is the result of greater diversity plus a steady and large night cleaning load. However, any tendency of our judgment to lean one way or the other should lean definitely toward the safe side. We cannot make water hot by wishing.

Miscellaneous Problems

Temperature Considerations. There is always the question of temperature. Restaurants must have 180 F water, if they are to win the approval of the board of health. For domestic service in homes, 140 F is said to avoid scalding, but a setting of 150 F may be needed at the heater to deliver hot water to distant faucets, as in the laundry. The 140 F setting also will suit employees of office buildings and banks, except possibly the night cleaners, who like their fluids as hot as practicable.

This sort of requirement can be satisfied in several ways. First, we can set up a base tank which heats all the water needed (both lavatory and dishwashing) to a uniform 140 F. From this base heater, the piping to the dishwashing facilities goes first through a booster heater, where another 40 deg is added to its temperature. This system will function quite satisfactorily if our lavatory load is a substantial one; otherwise it might be much better economically to use a 180 F heater and step down the temperature to the lavatories through an automatic mixing valve.

A second way of solving the problem is to make up the 180 F water in the dishwashing sinks by the use of the steam line described earlier in this chapter. Steam is generally available in a restaurant of any importance, since it is needed for the steam table, plate warmers, etc., and the diversion of a bit to the cleaning-up department is sometimes practiced by the trade.

Other arrangements will occur from time to time to those of us who become involved in restaurant problems.

Combination Systems. There is a problem that we might call the collaboration problem, which sometimes attempts to affect our estimates. It arises when there is a plan to use a domestic automatic water heater in collaboration with a furnace coil, or perhaps a stove. Unless conditions are quite special, sizing of the automatic unit should proceed as though the collaborator did not exist. The reason is that furnace coils and stoves are usually shut down in summer, and then the new heater is strictly on its own, and must meet all demands without help. The winter assistance of the collaborator is enjoyed by the owner in the form of reduced operating gas or electric costs on the heater; actually of course, he is paying out the difference for additional fuel in the furnace or stove.

Figure 7.10 illustrates a recommended way to hook up a furnace coil with an automatic heater. At times when the furnace is shut down, the old tank, which should be uninsulated at these times for best results, acts as an economizer, picking up warmth from the surrounding air and raising the temperature of the heater feed. During the coil operating season, the automatic heater will function as an automatic booster, merely floating on the line if the coil output is both hot enough and ample.

A frequent installation of the combination system is observed where the base load comes from a solar heater. No matter how sure and steady we hope the sunlight will be, there is always a possibility of days on end in which the sun fails to meet our demands.

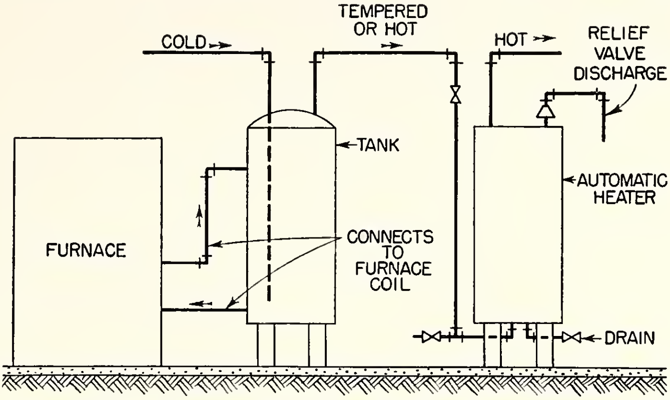


Fig. 7.10. Combination of furnace coil and automatic heater.

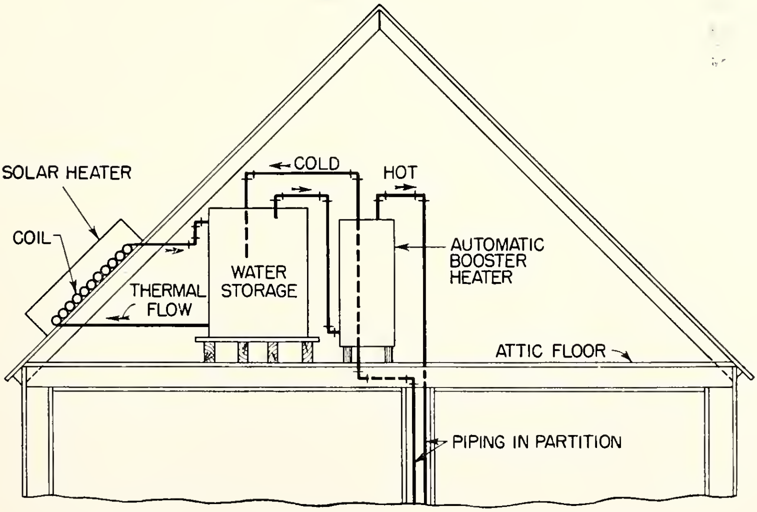


Fig. 7.11. Combination of solar and automatic heaters.

Figure 7.11 illustrates a method for using an electric heater in collaboration with the solar unit. When the water in the tank falls below, say, 130 F, the electric element is energized; at all other times it is dead. This system is admirable, but the cost of the supplementary heater may run the total investment to a higher figure than we care to accept.

Magnesium Rods for Corrosion. Nowadays, we need not often be troubled by having red and rusty hot water pour from our faucets. Rust in our water heaters is an electrochemical form of corrosion, accelerated by the high temperatures and movement of the water. Some of the shell areas act as the positive or anodic poles of a miniature battery, and the electric current which is discharged into the water at these areas is fed by the destruction of the metal behind it.

Corrosion is a complex and often puzzling study, but in water heaters much of it can be avoided by submerging magnesium rods in the water. These rods are more active electrically than is the steel of the shell, and they set up an electrical circuit with themselves acting as anodes. Such rods are really sacrificial rods, for in usurping the role of anode from the shell, they initiate their own destruction, and the shell is saved.

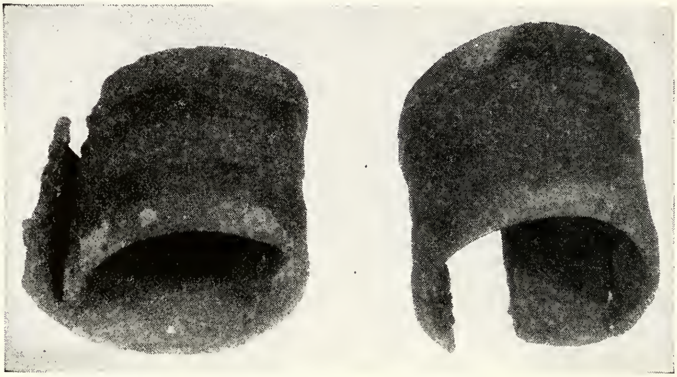


Fig. 7.12. Corrosion as it occurs in a boiler nipple.

Red water is a form of iron oxide originating at the point of corrosion when the affected metal is steel or iron, but magnesium does not foul our water during its process of destruction. To keep red water from our faucets, we need only install one or two magnesium rods in the heater, and we can count on protection lasting from one to many years, depending on the nature of the water and how often we use the heater. Many of the new heaters contain these magnesium rods as standard parts, while others are fitted for their easy installation at a later date.

Figure 7.12 shows what corrosion can do to ferrous metals.

Piping Arrangements. Two possible extravagances in the use of hot water are worth examining because of their influence on estimates:

1. If the piping runs are long, they should be insulated. This is particularly important if the piping traverses unheated spaces.

2. The use of a return system, while ensuring hot water at the faucets all the time and thereby eliminating the runoff of cold water, can increase operating costs up to 40 per cent. This increase is the result of our having to keep hot a large circulating bulk of water, whether we are using it or not.

Questions

1. Name the various types of water heaters and describe the dominant characteristic of each.
2. Assume that you are building a new home for your family. How would you go about choosing a water heater?
3. What safety precautions will you observe in installing *any* heater?
4. What size solar water heater would you choose for a family consisting of three adults, two children, and one full-time maid? Indicate the total hourly output in gallons, and give the minimum glass cover area necessary.
5. You are about to select a water heater for an apartment hotel having 150 tenants, of whom 15 are children under five years of age. There are 30 showers and 30 tubs, 10 slop sinks, and 45 lavatories. In the basement are 5 laundry tubs.
    a. What is the estimated peak demand, gallons per hour?
    b. Would you supply a storage tank? Why?
6. Calculate the hot water load of the drugstore at the corner which serves 125 counter meals with an average check of $1.10.
7. The drugstore in Question 6 has a public washroom with 2 lavatories, and 2 more lavatories for the employees. Water for dishwashing is 180 F. How much water at 140 F is needed for the lavatories? How would you provide it? Why?
8. You are trying to decide between a gas water heater and an electric water heater. The capacity of the heater is to be 40 gal per hr. Local gas rates are $1.75 per 1,000 cu ft of gas having a calorific value of 540 Btu per cu ft. Electric service will cost 6 cents per kilowatthour. Assume that each heater will operate 2,000 hr per year at full load and that the purchase prices are identical. Which is the cheaper to own over a period of 10 years?
9. You plan to heat water for an office building using steam from district steam street mains. The load is intermittent with a peak of 500 gal per hr between 12 and 1 P.M. The average hourly demand is only 90 gal per hr. If steam demand charges are $1.25 per pound of maximum demand, how large a storage tank would you install?
10. Can you use a building heating boiler as a hot water generator? How?

CHAPTER 8

DESIGNING A WARM AIR HEATING SYSTEM

Engineering progress is a curious activity of mankind, not infrequently occurring as the result of a reversal of thought. For a case in point, observe what has happened to our opinions of warm air heating. This was probably man's first improvement over the open fire. Many of us can remember the registers in the walls and floors that exhaled warm comfort on frosty days, enlivened occasionally by dust explosions in the heater or ducts that painted the rooms with dirt and smoke.

These warm air systems with their explosions operated simply because the reduced density of warm air caused it to rise; delivery in consequence was not always a sure thing in every room. Long horizontal runs of duct sometimes so nullified the rising force by friction and cooling that the rooms at the ends of these ducts were consistently uncomfortable in cold weather.

These characteristics of warm air heating impressed on it a very dubious reputation. People who really insisted on their comfort chose steam or hot water without hesitation, and warm air systems all but disappeared from new construction.

Then came air conditioning. The old ducts were dusted off, and suddenly everybody realized that a fan which could make a register blow cold could as readily make it blow hot. The compulsion of the fan also could force the air through a filter, thereby eliminating the explosions and filth so characteristic of the old days; a pan of water could add desirable humidity; violet-ray lamps in the air stream could destroy air-borne bacteria; glycol solutions introduced as a spray could attack the virus of the common cold; in short, here was an atmosphere for comfortable living literally made to order.

Warm air heating, new style, is now being accepted in our best circles, and with a celerity that is hardly less startling than was one of its ancient explosions. The familiar cycle of an idea discarded and then exhumed and improved upon is completed anew. Engineering moves on.

Designing the Gravity System

Most of us never will design a gravity system, but an understanding of how it is done definitely widens our understanding and skill as applied to the entire field of warm air heating.

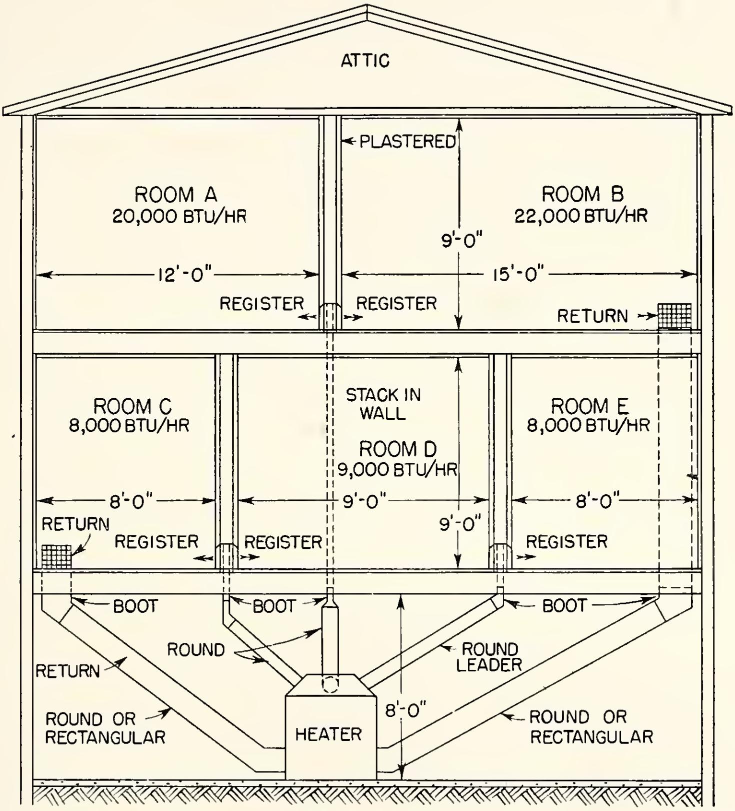


Fig. 8.1. Two-story house and basement, heated by gravity warm air system.

Figure 8.1 is a sectional elevation of a two-story house with its basement. In designing a gravity warm air system for this house, we shall work with the following basic assumptions:

1. The register discharge temperature is 175 F. This is the normally recommended temperature.
2. Cross-sectional areas of the wall stacks shall be not less than 75 per cent of the areas of the leader pipes that supply them.
3. The pitch of leaders from heater to stacks shall be at least 1 in. per ft.

4. Registers shall be placed low on the side walls, or in the baseboard, rather than in the floor or high on the side walls. High wall gravity registers with their limited outlet velocity tend to develop hot pockets close to the ceiling. Floor registers are dirt sumps and fire hazards.

5. Clear areas in the registers shall be not less than the cross-sectional area of the supply leader in the basement.

6. Return grilles will be placed close to the floor, with their top edges not higher than 14 in. above the floor. Additionally, we shall choose a site in, or near, the coldest wall or largest exposure of window glass.

7. While one or two return connections usually is all we need, we will provide an individual return for any room with a floor level below the surrounding floors. Otherwise, we would have a cold pocket in this lower room.

8. The cross-sectional area of the return duct or ducts must equal the sum of the cross-sectional areas of the supply leaders.

9. The downward slope of the return piping in the basement shall be at least $\frac{1}{2}$ in. per ft. If we can achieve a slope of 45 deg, that is ideal.

10. Return air temperature is assumed to be 70 F. If the temperature is lower than 70 F, then we shall have a greater differential of density to augment our circulation. Higher temperatures of course have the reverse effect, and slow down air movement.

Sizing the Warm Air Leaders and Stacks. The National Warm Air Heating and Air Conditioning Association has accomplished a good and sizable job in developing dependable capacities for various sizes of piping in gravity warm air systems. The Code and Manual for Gravity Systems, published by this association, should be on the shelf of every engineer and contractor who is in any way concerned with gravity air heating. All the short cuts of design are there for our benefit.

The Engineering Experiment Station of the University of Illinois also has developed a fast and dependable approach to the problem. In either case, a willing ally stands behind our pencils. We need to guess about nothing.

To size up the system for the house in Fig. 8.1, we consider each room in turn. Room *A* has a leader approximately 8 ft long, one 90-deg elbow or transition where it turns upward into the wall stack, and a total load of 20,000 Btu per hr. On referring these data to Table 8.1, we note that the load is too large for a single delivery; we must heat room *A* with two registers served by two stacks.

Dividing the load into equal components of 10,000 Btu each enables us to supply room *A* by means of two 10-in. leaders with 14 × $3\frac{1}{4}$-in. stacks. This gives us a nicely balanced arrangement of piping sizes, but we can save a little money by using only one 10-in. leader, supplemented by one 9-in. leader with a 12- by 3¼-in. stack. According to the table, the 10-in. leader will deliver 11,050 Btu, the 9-in. leader 9470 Btu, making a total of 20,520 Btu.

TABLE 8.1. LEADER CAPACITIES, BTU PER HOUR

No. of elbows allowed	Leader diam., in.	Length of leader, ft			
		8	12	16	20
1*	8	5,680	5,340	5,000	4,660
	9	7,180	6,760	6,320	5,890
	10	8,870	8,340	7,810	7,270
	12	12,590	11,830	11,080	10,320
	14	16,530	15,550	14,550	13,560
2*	8	5,490	5,160	4,840	4,510
	9	6,940	6,520	6,110	5,700
	10	8,580	8,060	7,550	7,040
	12	12,170	11,430	10,690	9,950
	14	15,990	15,040	14,080	13,120
3*	8	5,310	4,990	4,670	4,350
	9	6,710	6,310	5,900	5,500
	10	8,280	7,780	7,290	6,800
	12	11,750	11,050	10,350	9,650
	14	15,440	14,510	13,600	12,660
1†	8	7,900	7,430	6,950	6,470
	9	9,470	8,900	8,330	7,770
	10	11,050	10,390	9,720	9,060
	12	15,300	14,380	13,460	12,550
2†	8	7,500	7,050	6,600	6,150
	9	9,000	8,460	7,920	7,380
	10	10,500	9,870	9,230	8,610
	12	14,530	13,670	12,800	11,930
3†	8	7,110	6,680	6,250	5,830
	9	8,520	8,010	7,500	6,990
	10	9,940	9,350	8,750	8,160
	12	13,780	12,950	12,120	11,300

* Includes register and stack to first floor.
† Includes register and stack to second floor.

The supply to each of the rooms is sized in the same way. A summary of the warm air piping is given in Table 8.2.

Sizing the Return Ducts. Our next step is to size the returns. Two return connections are shown for this house, one on the second floor, one on the first. Probably we could recover all the air we need by a single intake located in the vicinity of the stairway, for nature decrees that cold air shall come tumbing down a stairway if the air on the lower floor is warm. In this case, however, the owner might convert his building into an apartment house, and the complications arising from the likes and dislikes of tenants one for the other suggest separate returns as being simple prudence. Whatever else happens, the lower tenants will not be subjected to showers of cold air from the upper.

Table 8.2. Summary of Gravity Warm-air Supply Piping

Room	Total load, Btu/hr	Length of leader, ft	Equivalent elbows	Leaders			Stack size, in.
				No.	Diam., in.	Btu	
A	20,000	8	1	1	9	9,470	12 × 3¼
				1	10	11,050	14 × 3¼
B	22,000	8	1	2	10	11,050	14 × 3¼
C	8,000	6*	1	1	10	9,135	14 × 3¼
D	9,000	10*	2	1	12	11,800	
E	8,000	12	2	1	12	8,060	

* *Note:* For these lengths we must exterpolate in the case of room *C*, and interpolate for room *D*. In the former, the average change per foot of length is

$$\frac{8{,}870 - 8{,}340}{4} = 132.5.$$

For the latter $\frac{12{,}170 - 11.430}{4} = 185$. The capacity for room *C* is therefore 8,870 + (2 × 132.5), or 9,135 Btu per hr. Room *D* being halfway between the 8- and 12-ft lengths of leader will deliver 12,170 − (2 × 185) or 11,800 Btu per hr. Capacity decreases as leader length grows.

A basic rule for sizing returns is this: The cross-sectional area of the return piping should equal the total cross-sectional area of all the supply leaders. However, the diameter of the leader to any or all of the rooms actually may be a little larger than is necessary, owing to the spread between the sizes as shown on Table 8.1. For example, if we had a load of say 10,000 Btu per hr to be met by a leader 8 ft long and with only one elbow, the table indicates that a 10-in. pipe is too small and that we must use a 12-in. leader having a capacity of at least 20 per cent above our requirements. If this situation occurs two or three times in a single job, which is quite possible, a return based on the sum of the 12-in. pipe areas will be larger than we need. The safest procedure is for us to keep the basic rule in our minds, but to take our return pipe sizes from Table 8.3, which has been computed in terms of Btu capacity.

Our second-floor load totals 42,000 Btu per hr, and with the most favorable arrangement possible we shall need a 20-in. duct. For the first floor, we have a total of 25,000 Btu per hr, and the duct required to keep the cold air moving back to the heater must be 16 in. in diameter.

Let us see how the areas of these return ducts compare with the summation of the supply areas.

TABLE 8.3. RETURN DUCT CAPACITIES, BTU PER HOUR

Duct sizes, in.		Area, round, sq in.*	Size of intake grille, in.	Capacity		
Rectangular	Round			Round	Rectangular	With lined joists
14 × 6	10	86	8 × 14	11,300	5,000	7,800
22 × 6	12	124	6 × 30	16,300	7,200	11,300
28 × 6	14	170	8 × 30	22,200	9,800	15,300
28 × 8	16	220	10 × 30	29,000	12,800	20,000
38 × 8	18	280	12 × 30	36,700	16,200	25,300
36 × 10	20	340	14 × 30	45,300	20,000	31,300
42 × 10	22	420	18 × 30	54,800	24,100	37,800
42 × 12	24	500	20 × 30	65,200	28,700	45,000

* Areas listed are equal to approximately 110 per cent of the areas of the circles having the diameters shown.

The return that is 20 in. in diameter provides a cross-sectional area of 314 sq in.; the 16-in. return, 201 sq in.

Supply piping to the second floor comprises one 9-in. pipe of 64 sq in. area, and three 10-in. pipes worth 78 sq in. apiece. The supply total is therefore 298 sq in., and our 20-in. return generously exceeds our demands by 16 sq in.

To serve the first floor, there are one 10-in. and two 12-in. pipes. According to Table 8.3, which pleasantly saves us the trouble of calculating the areas of circles, these three pipes add up in total area to 334 sq in. Since our return provides only 201 sq in., the situation seems to be quite out of balance. Actually, the supplies are oversized, and our 16-in. return will perform most satisfactorily. This case demonstrates the clear advantage of sizing returns on the Btu basis rather than by equal areas.

The most favorable return design is shown by Fig. 8.1. Here the duct falls swiftly to the heater shoe; there are no horizontal runs, no offsets, nothing to introduce any out-of-the-ordinary resistance. This is what we shall do when we can. If we must employ a more tortuous connection, we shall discount the capacities shown in Table 8.3 anywhere from 10 to 25 per cent, depending on the complexities of the design.

Exact reductions can be obtained from the Code and Manual for Gravity Systems, tabulated for numerous arrangements of the return connections.

Returns in the Joists. To conserve on space and materials, returns in the basement frequently are confined between the joists of the first floor. A very cheap installation is obtained by the simple act of nailing flat metal sheets across the open space between parallel joists. This sheet becomes the bottom of the duct, the joists provide the walls, and the underside of the first floor forms the roof.

Obviously there is no pitch to this sort of duct, and as the rough surfaces of the joists and underside of the floor chasten the easy passage of the air, expectations of capacity must be modest to say the least. Often, to obtain the necessary cross-sectional area, we shall need to spread the return over three joists. For instance, with 14 in. clear space between joists 12 in. deep, the available cross-sectional area is only 14 × 12 = 168 sq in. This is 8 sq in. less than the area of a 15-in. pipe.

As a general basis for estimating, we should deduct 20 per cent from the return capacities shown in Table 8.3 if we decide to employ the joists as duct walls. If the run is very long, say over 10 ft, wisdom warns us that 25 per cent is even better, with 30 per cent to be considered under particularly unfavorable arrangements. Table 8.4 gives dimensions for joist ducts.

Table 8.4 Dimensions of Lined Joist Ducts for Return Air*

No. of Joists with Lining	*Minimum Depth, In.*
1	7
1	9
1	12
2	8
2	10
2	12.5–18

* *Note:* Abstracted with permission from the 1948 ASHVE Guide.

These minimum depths are based on a clear space between the joists of 14 in. When the joist depth is less than the minimum, a pan must be attached to the bottom of the joists, to provide the space needed.

There is now developing a tendency to eliminate return ducts in the basement. In this scheme, the return stacks simply terminate under the first-floor joists, and the air finds its own way back to the heater.

While present applications appear restricted to gravity systems, undoubtedly we may expect to use it under favorable conditions, for the mechanical designs as well. The basement functions as a huge plenum chamber on the return side of the heater, whether the system is mechanically or gravity actuated, and, if basement doors and windows are kept closed, satisfactory results may be expected.

An Alternate Procedure for Sizing Leaders. In Table 8.5 are figures showing the heat-carrying capacity of leaders, per square inch of cross-sectional area. This table is based on a return temperature of 70 F, the supply temperatures to be as we choose. For example, the conventional 175 F supply will deliver 111 Btu per sq in. of pipe area to the first floor; 167 Btu to the second floor, and 200 Btu to the third floor.

In reference to room *A* in Fig. 8.1, which requires 20,000 Btu per hr and which is located on the second floor, the cross-sectional area necessary to deliver this quantity of heat is 20,000/167 = 120 sq in. A 12-in. pipe offers an area of 113 sq in., which is too small; therefore we go to the next size, 13 in., and will obtain an area of 132 sq in.

Compared to pipe sizes selected from Table 8.1 for room *A*, namely, one 9 and one 10-in. pipe with a total area of 142 sq in., this alternate procedure seems to produce results slightly on the skimpy side. However, the values in Table 8.5 have been computed from data obtained in numerous field tests, and actually are carefully weighted averages. So many factors can affect the operation of a gravity system, such as prevailing barometer for example, that the data collected on one day may not be duplicated for a week.

TABLE 8.5. LEADER CAPACITIES, BTU PER SQ FT, OF CROSS-SECTIONAL AREA*

Air temp., F	Floor being served		
	1	2	3
140	52	96	118
150	70	118	143
160	80	140	166
170	95	155	192
175	111	167	200

* These values are approximate and have been computed from data in the ASHVE Guide, 1942 Edition.

Do not use this table or any design in which the leaders are longer than 8 ft.

Since these variations in performance are normal for every system, no matter how computed, the difference between the two methods of sizing is of less importance than it seems. The alternate method is quite safe, and we may use it as we wish. In fact, by remembering the basic values of 111, 167, and 200 Btu, respectively, fast estimates of sizes and prices are possible for many jobs.

Wall Stacks. The standard partition is formed of wooden studding nominally 2 by 4 in. in size, set vertically on 16-in. centers.

The actual depth of these timbers, however, is more nearly 3½ in. than 4 in., due to the planing which serves to finish their surfaces smoothly.

This situation limits the maximum dimensions of our stack ducts in normal construction to 14 in. wide and 3½ in. deep. In fact 3¼ in. is considered standard depth for stack practice.

Architects will permit us to use a 5¼-in. dimension at times, if the design of the building structure permits the employment of a partition 6 in. deep. More often, if a large stack area is needed, it is easier to widen the stack beyond the 14 in. of conventional maximum and to frame the partition around it accordingly.

The definite limitations of space available to wall stacks must be kept in mind at all times. The installation of two stacks may be expected to cost more than the installation of one, but occasions will be met that leave us no other choice. When we cannot change the partition framing either in depth or width, double stacks must be used. One consolation is that for a load so large that we cannot serve it through a standard spacing two registers will achieve a much better distribution of heat at the points of delivery.

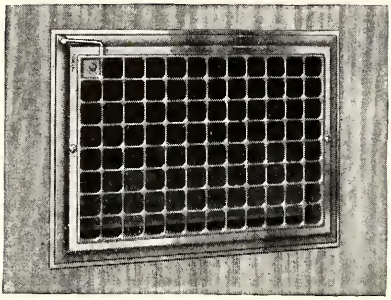


Fig. 8.2. Gravity type of register. *(Courtesy of Hart & Cooley Manufacturing Company.)*

Table 8.6. Resistance of Boot Forms, in Equivalent 90-deg Elbows*

Form of Boot	*Equivalent* 90-*deg Elbows*
45-deg angle plus 45-deg elbow	1
90-deg angle	1
Universal plus 90-deg elbow	1
End boot	2
Offset boot	2½
Floor register	3
Offset, rectangular in cross section	3
Offset, round in cross section	2
45-deg elbow	½

* *Note:* These data have been abstracted with permission from the 1948 ASHVE Guide. This table should be coordinated with Table 8.1 in computing the carrying capacities of ducts.

The Registers. The most important rule to remember in selecting gravity registers is this: The free area of the register must be not less than the free area of the leader which supplies it.

Figure 8.2 illustrates a type of gravity register that is familiar to all of us. Let us consult Table 8.7, which provides us with the physical data

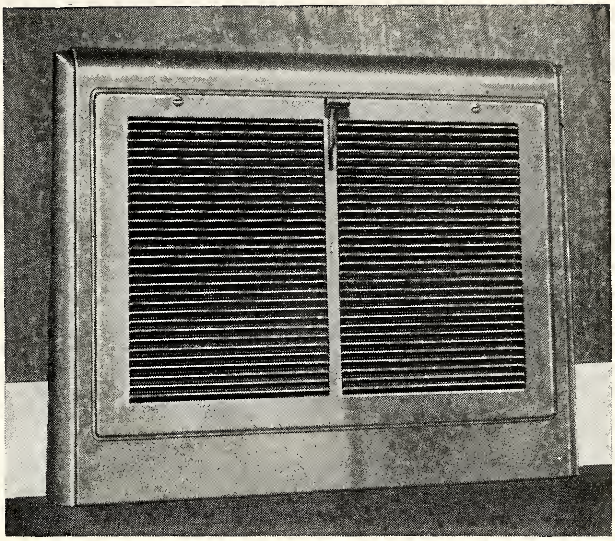


Fig. 8.3. Fin-type wall register. *(Courtesy of Hart & Cooley Manufacturing Company.*

on this type of register, and pick out the registers for the house in Fig. 8.1.

TABLE 8.7. DIMENSIONS OF GRAVITY SIDE-WALL REGISTERS

Size of stack head opening, horizontal* dimension first, in.	Over-all size of register face, in.	Free area, sq in.
8 × 10	10¼ × 12¼	56
8 × 12	10¼ × 14¼	67
9 × 12	11½ × 14³⁄₁₆	75
10 × 12	12³⁄₁₆ × 14³⁄₁₆	84
12 × 14	14⅛ × 16⅛	118

* Any of these registers may be set vertically or horizontally as desired.

SUMMARY OF APPLICATIONS

Size of leader, in.	Area of leader, sq in.	Register	
		Size, in.	Area, sq in.
9	63.6	8 × 12	67
10	78.5	10 × 12	84
12	113.0	12 × 14	118

New register designs have been developed in recent years which not only afford us a much better looking appearance but yield greater free area as well.

Figure 8.3 illustrates one of these adjustable fin designs.

Observe how the naked background of the stack walls is concealed. We are not compelled, as in the register of Fig. 8.2, to stare at the unlovely appearance of the machinery that delivers our comfort. In fact, we can close the opening to any degree we wish, simply by tilting the adjustable fins.

Table 8.8. Dimensions of Fin-type Gravity Registers

Size of stack head opening, in. horizontal dimension first	Over-all size of register face, in.	Free area sq. in.	Projection of face, in.
10 × 8	13⅛ × 12	63	2¼
12 × 8	15⅛ × 12	78	2¼
12 × 9	15⅛ × 13	88	2¼
12 × 10	15⅛ × 13⅞	98	3¼
13 × 11	16⅛ × 14⅞	118	3¼

Now let us examine the free area of this design (see Table 8.8). The advantages seem to be quite obvious. However, nothing is ever perfect, and we should consider its limitations as well as its virtues. First, this one is to be installed with the horizontal dimensions as given. We cannot turn it vertical, on end. Second, it projects a few inches beyond the wall, or baseboard. Third, it costs approximately one-third more.

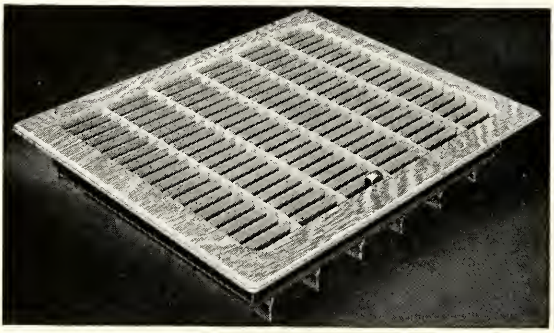


Fig. 8.4. Floor register. *(Courtesy of Hart & Cooley Manufacturing Company.)*

Shall we adopt it? Undoubtedly many fine residence jobs will accept nothing less. On the other hand, many modest owners, to whom the expenditure of every penny is important, will be quite content with the cheaper register.

The floor registers are illustrated in Fig. 8.4 (for dimensions see Table 8.9). This type of register may be used in the ceiling or side wall also. It is built of heavy steel, capable of carrying the weight of many persons who walk across it in a day. Under the grating face are louvers that may be opened and closed at will.

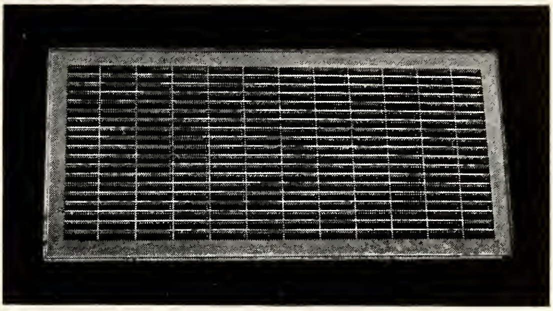


Fig. 8.5. Return grille for wall installation. *(Courtesy of Hart & Cooley Manufacturing Company.)*

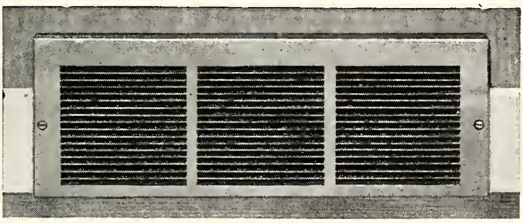


Fig. 8.6. Baseboard return grille. *(Courtesy of Hart & Cooley Manufacturing Company.*

TABLE 8.9 DIMENSIONS OF FLOOR REGISTERS

Body size to fit opening, in.*	Over-all sizes of face, in.	Free area, sq in.
16 × 16	18⅜ × 18⅜	190
16 × 18	18⅜ × 20⅜	213
16 × 24	18⅜ × 26⅜	284
18 × 24	20⅜ × 26⅜	319
20 × 24	22⅜ × 26⅜	355
30 × 30	32⅜ × 32⅜	674

* Dimensions given do not cover the entire variety of sizes available. Manufacturers' catalogues should be consulted for the complete list.

The return grille for a gravity system normally is without louvers or fins. Figure 8.5 shows a typical design, intended for installation in a wall. More than 80 per cent will be free area in this grille.

Figure 8.6 shows the type of return inlet we see frequently in homes, hotel rooms, and similar residential environments. It is built for installation in a baseboard and has flexible fins, or louvers, which enhance the appearance. The free area is a little restricted, however. For instance, the 10- by 8-in. size provides only 59 sq in.; the 12- by 8-in. unit, 72 sq in. Compare these free areas with similar-sized registers on the supply side, and the restriction of area is material.

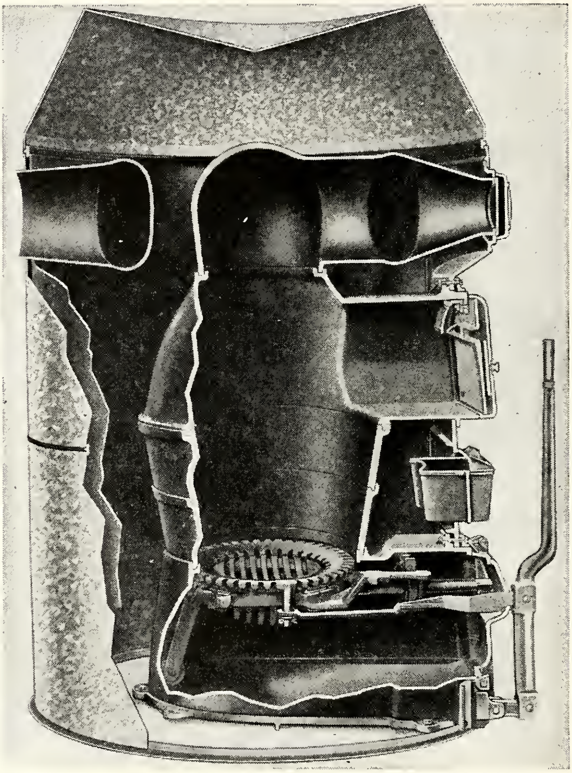


Fig. 8.7. Gravity type of warm air furnace, using cast-iron heater body and galvanized steel casing. *(Courtesy of American Radiator and Standard Sanitary Corporation.)*

Characteristics of the Gravity Furnaces

The most simple design of the gravity furnace which is used in duct service is illustrated by Fig. 8.7. Here we have what is really an adaptation of the old, familiar, pot-bellied stove. The cast-iron combustion space is enclosed in a steel jacket, the air moves upward and around the cast-iron heart and eventually reaches the distributing bonnet, or hood, at the very top of the heater. From this point, the warm air issues to the room of its ultimate destination by way of the ducts.

Figure 8.8 shows us a variation of the cast-iron design; the combustion chamber is formed of riveted or welded steel and is lined with refractory brick. Thus we have an improvement, but like all improvements, we must pay for it; the steel furnace, in short, costs more money.

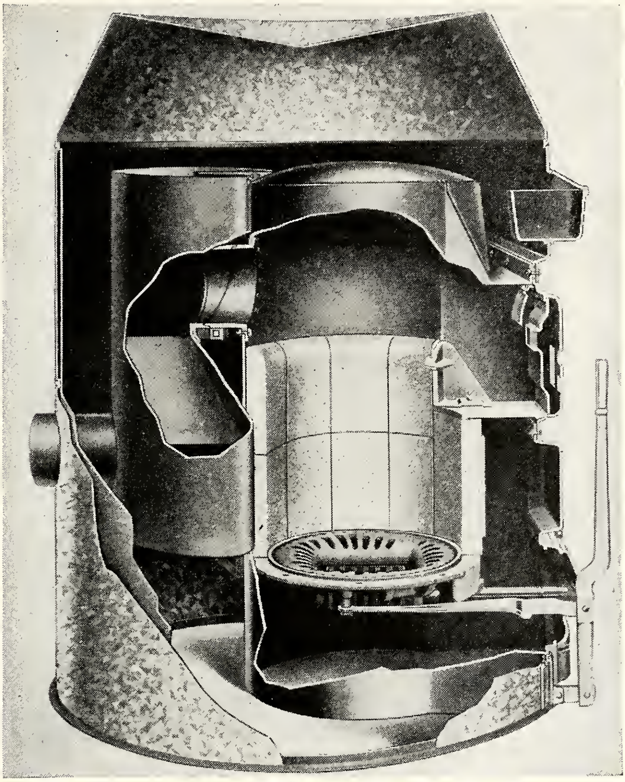


Fig. 8.8. New steel gravity furnace. *(Courtesy of American Radiator and Standard Sanitary Corporation.)*

Any of these furnaces may be obtained for firing coal, coke, or oil, and the solid fuel may be fired by hand or by stoker. There is considerable significance to the manner of firing; the fuel affects the rating. For example, consider the ratings in Table 8.10 for various fuels, as offered by a single size of furnace. Obviously, in selecting a furnace, it is necessary to know what kind of fuel will be used.

TABLE 8.10

Fuel	Rating, sq in.	Rating, Btu/hr
Soft coal	1,172	159,400
Hard coal or coke	938	127,600
Oil	1,346	183,050
Stoker	1,476	200,700

Conversion of Ratings. The capacity of a furnace in terms of Btu is a very handy bit of information to have, since this knowledge permits us to evaluate the suitability of the furnace to the number of floors we must heat.

For example, suppose that we refer to a manufacturer's catalogue and pick out a unit rated at 150,000 Btu when fired with oil. For a bungalow, with all the heat being delivered to the first floor, our leader capacity would be 150,000/111 = 1,361 sq in., approximately. For the second floor exclusively of an apartment house or flat type of dwelling, these 150,000 Btu/167 = 900 sq in.; if we had to deliver the heat to the third floor, our leader area would be 150,000/200 = 750 sq in.

Now suppose that we reverse the procedure and convert square inches into Btu. For our two-story house (Fig. 8.1), the way we shall select the heater is illustrated in Table 8.11. The computed heat load is the actual heat loss that must be met from the furnace delivery. It consists of 42,000 Btu on the second floor, 25,000 on the first.

TABLE 8.11

Floor	Total leader area, sq in.	Capacity, Btu/sq in.	Total leader capacity, Btu/hr
First	304	111	33,744
Second	298	167	49,766
Total capacity	...	...	83,510
Computed heat load (Fig. 8.1)	...	...	67,000

Assuming that we shall fire with oil, we refer to a manufacturer's catalogue and observe what is available. Any unit capable of delivering at least 67,000 Btu per hr will be satisfactory, and there is no need to choose one for more than 83,510 Btu capacity, because that latter value is the limit for the ducts.

There is one cautious thought we should keep in mind when selecting a heater: If there is any likelihood of the owner wanting to change to another fuel at some later date, we should size the heater for the lower rated fuel, and not the higher. Thus, a heater sized for oil may become inadequate if changed over to hard coal. The safe plan is to size the heater for coal; then a later conversion to oil could be achieved satisfactorily. However, the owner should be informed of this situation, for sizing on the lower rated fuel may involve a larger furnace at the start, and consequently a larger investment.

Dampers. A hand operable damper must be provided in each of the leaders save one. The nondampered leader should be chosen to supply a space that is heated most of the time. In short, this is the "base load leader," and the system is balanced on this leader.

The undampered leader performs another important function. It is our watchdog of safety. If all the dampers in all the leaders were closed, and if we had no wide open pipe to permit the escape from the furnace of the heated air, overheating of the furnace, distortion of the metal, and perhaps a fire might result. As it is, overheating in the room served by the nondampered leader soon impels us to check the fire in the furnace.

Of course, the application of automatic controls can lift much of this responsibility from our shoulders. A temperature-limit switch, with a thermostat in the bonnet, quickly shuts off the furnace draft, stops the stoker, extinguishes the flame of the oil burner, and thus effectively arrests combustion when the bonnet temperature exceeds the setting of the switch. Automatic controls add to the cost of the installation, and since the strongest recommendation for a gravity system is its low investment, these controls seldom are purchased. Most owners prefer to assume hand control.

Balancing a Gravity System. Balancing a system can be accomplished in two ways. First, we set the dampers by guess, and then either estimate the comfortable feeling of a room or check temperatures with a thermometer. Since temperatures will not be uniform (indeed they may vary by 10 deg or more just between the floor and ceiling), the story told to us by the thermometer is not too significant. Individuals also react differently to ambient temperatures, and what seems to be perfect comfort to one may confer nothing but chilliness on another. This fact may initiate a period of damper juggling if we are not careful.

The proper way to start balancing a system is to assign two men to the task, assisted by one velometer. By reading the discharge volumes from each register on the velometer, the designed output can be set by the man at the damper.

We go this way from room to room, and when the rounds are completed, the heating system is balanced as it should be, theoretically. Generally, not much changing of damper settings will be needed for a system so balanced; here and there a room may feel overheated or underheated, but the adjustment is made easily, for the variation from the theoretical will not be wide.

Converting Btu into Cfm. To use a velometer, we must know how to convert the room requirements in Btu to cubic feet of air per minute (cfm). This is done without trouble as follows:

For room *A*, in Fig. 8.1, we have a load of 20,000 Btu per hr. Divide this load by 60, and the answer is in Btu per minute, 334 approximately.

Now the warm air reaches the room at 175 F and leaves near the floor at

70 F. The difference is 105 deg of temperature. Since the specific heat of air is 0.24, every *pound* of this air that enters the room hot and leaves cool will unburden itself of 105 × 0.24, or 25.2 Btu.

How many pounds of this air is needed in order to have 334 Btu per min? Just $334/25.2 = 13.25$ lb, approximately.

Now we refer to Table 2.7, which provides us with the properties of air, and observe that 175 F air will occupy about 16 cu ft per lb.

Multiply 13.25 by 16, and the answer is 212 cfm. Set the damper so that the register discharge velocity as read on the velometer is $\frac{212}{\text{Free Area of Reg., Sq. Ft.}}$ and room *A* is being assigned the heat it needs.

Figure 8.9 pictures a velometer. Alternately an anemometer may be used for balancing, but the results are likely to be less accurate.

Insulation of Leaders and Ducts. In common practice, warm leaders are insulated if they exceed 8 ft in length. Actually a better practice is to insulate all the leaders, whatever their length. We depend on the difference of air densities to circulate the air throughout the system, and every degree of temperature lost in the leaders detracts from the force behind this circulation.

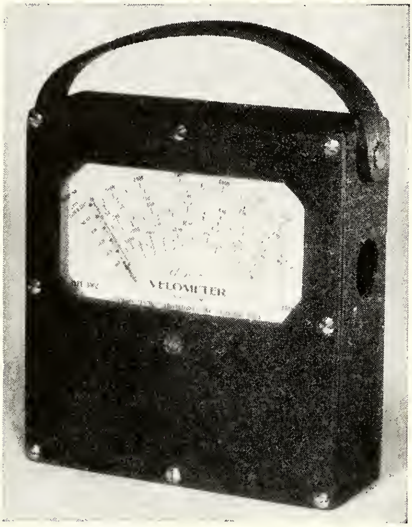


Fig. 8.9. The velometer for measuring air velocities. (*Courtesy of Illinois Testing Laboratories, Inc.*)

Conceivably, bare leaders may be desired in some cases to heat the spaces through which they pass. In these cases we should plan on burning a little more fuel, to make up for the leader losses and to maintain high warm air temperatures.

Insulation with air-cell asbestos, at least ¼ in. thick, is recommended for all leaders that happen to be located in areas of normal exposure, for instance in the average basement. If outside air can reach the pipe, or if it is subjected to cold drafts, the thickness of the insulation should be doubled. Occasionally, pipes must be run through sheds where there is little or no siding and the only protection is from a roof; in these cases the double-thick insulation should be wrapped in a coat of asphalt roofing, securely wired on.

Return ducts are not covered at all, unless exposed to the weather.

The reason for this is that if the air within them grows a little cooler in its journey to the heater circulation is augmented by the increased density so developed.

Chimneys for Gravity Heaters. There are two ways of sizing a chimney. One way calls for the exercise of mathematics in the evaluation of drafts, friction, and the products of combustion. This procedure is discussed in detail in Chap. 16.

The other way is far simpler. We ask the builder of the heater how big a flue he needs and how high a chimney. Many of the catalogues already contain these data for each size and model of furnace. Just select the proper heater, and the chimney data will be given with it. As an example, the data in Table 8.12 have been abstracted from a catalogue.

TABLE 8.12

Capacity of heater, Btu/hr	Flue size, in.	Height of chimney, ft*
61,000	8 × 8	25
74,000	8 × 8	25
90,000	8 × 8	30
105,400	8 × 12	30
125,900	8 × 12	30
183,000	12 × 12	35

* Height measured from top of grate.

Manufacturers are very careful to specify the proper chimneys, since the satisfactory performance of their furnaces can be prevented by an unsuitable chimney.

Mechanical Circulation Systems

Figure 8.10 is exactly the same as Fig. 8.1, except that we are now using a forced circulation heater, instead of a gravity heater. This similarity of subjects has been chosen with the idea of pointing up more sharply the differences between the two systems.

Procedure for designing a forced circulation system is as follows:

1. Determine the heat losses from each room. In this case assumed values have been indicated on the drawing for each room.

2. Convert the Btu per hour, for each room, into cubic feet per minute of air. The method for this conversion was described in the section on balancing a gravity system. At this point, we must make a decision, namely, how much excess capacity to allow for heating up periods and intermittent operation. Buildings are to Btu what sponges are to water. If the building has become chilled, warming the walls may require con-

siderable time and use up more Btu than we have available if we are not careful.

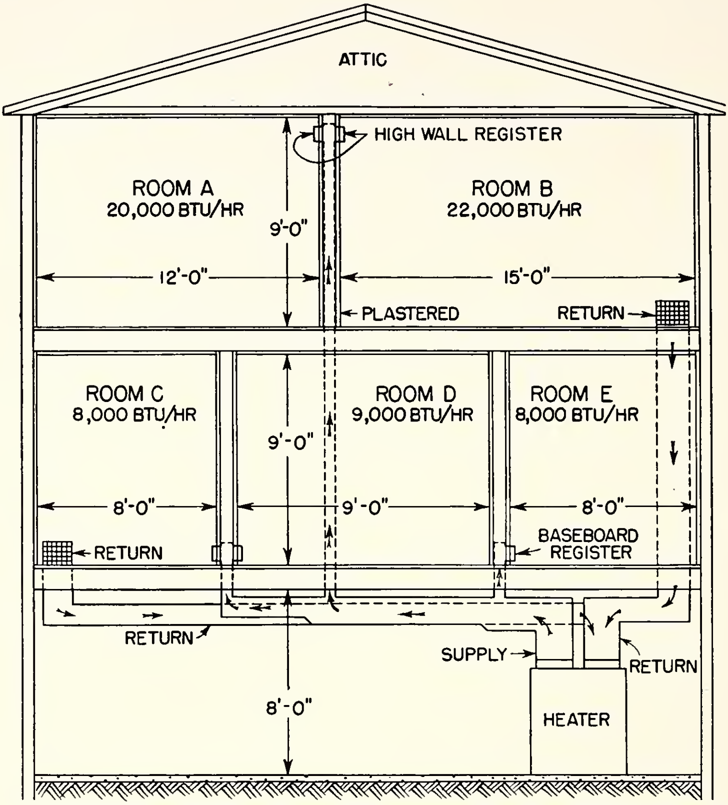


Fig. 8.10. Two-story and basement house arranged for mechanical warm air heating.

Opinions differ as to the excess we should allow. However, the requirements are clearly a function of the outside weather, and the following schedule is suggested:

a. For mild climates, with an average winter low of 20 F, add 25 per cent.

b. For design lows of 0 F, add 50 per cent.

c. If −15 F is expected, add 75 per cent.

d. For anything more severe, add 100 per cent.

These allowances should be increased in areas of high winds in accordance with our evaluation of the specific conditions. Winds are important because their movement across the surfaces of a structure multiply the convection losses substantially. In our problem example, suppose we add 50 per cent to the calculations of cubic feet per minute, to cover all factors.

3. Lay out a line diagram as illustrated in Fig. 8.11. Alternately, we may draw in the ducts from the heater to each register, but the diagram, during the sizing period, seems to clarify the problem. Then when the ducts are drawn in the final design, they may be drawn to scale, and erasures and changes are avoided on the finished drawing.

4. Determine the size of each duct and branch. Basing the sizes on velocities is apparently the simplest procedure, and since the results will be quite satisfactory, this is the method we shall use. Some designers prefer to size their ducts on the basis of equal friction. However, the computations are more complex, and most of us tend to shy away from such intricacies. Engineering is hard enough at its simplest.

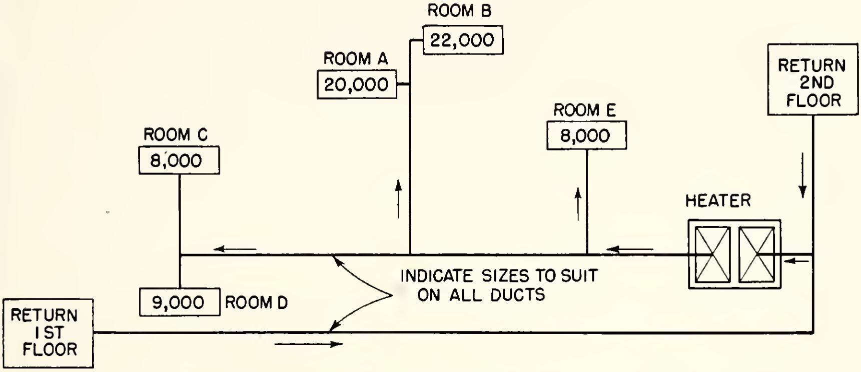


Fig. 8.11. Line diagram for sizing a mechanical warm air heating system.

Here are velocities that have been found to perform very well in practice:

Main ducts	800–1,000 fpm
Branches	600– 700 fpm
Wall stacks	500– 600 fpm
Main riser trunks	500– 600 fpm
Judgment factor	Favor the lower velocities as the distance from the heater increases. Pressures are falling in these distant ducts, and large losses from friction cannot be accepted

5. Size the returns similarly. With a fan in the system, we shall convert our returns into cubic feet per minute at the return temperature, and thereby save a little on the size of the ducts. Exactly how this is done will appear a little later on, when we demonstrate these procedures with figures.

6. Select the heater. Warm air practice indicates that we should expect a 20 per cent difference between the bonnet rating of a heater and the register rating. Thus, 100,000 Btu at the bonnet may be expected to deliver only 80,000 at the registers.

7. Select the registers. The product of each manufacturer is designed to meet specific conditions of volume, direction, and "throw." The term "throw" is given to the distance, in feet, that a stream of warm air may be expected to travel on emerging from a register. Throw is important for this reason: If we set a register high on an inside wall of a room, so that it faces an outside wall 20 ft away, we must somehow project the warm air across that 20 ft if the area is to be heated. Each register has an inherent capacity for "throw," as designed by the manufacturer. In selecting the registers, we must choose them not only for volume discharge but also for their ability to project this discharge the necessary distance.

8. Decide on the control of the heating system. Most residential installations are governed adequately by a single thermostat. However, the controls for a large structure are more complex, frequently involving zone action, motor-operated dampers, and thermally controlled registers for individual rooms.

Figuring for a Specific Building. Having looked at the design procedure in general, suppose that we now implement the individual precepts with figures.

Our first step is to convert the Btu loadings for each room in Fig. 8.11 into cubic feet of air per minute. With a register temperature of 150 F and a return temperature of 70 F, both commonly accepted values in heating practice, the harvest of Btu from each pound of air delivered is 19.2.

We saw how this conversion was accomplished in the section under gravity system design; however, when a number of rooms are involved we can save ourselves some mathematical repetition by simplifying the basic formula:

$$\frac{\text{Load per hour, Btu} \times 15.4}{60 \times 19.2} = \text{cfm}$$

In this formula, the 15.4 is the volume of air at 150 F (from Table 2.7); the 60 is for minutes in an hour, and of course the 19.2 represents the Btu available per pound, under these conditions.

Now we multiply and divide all we can with the known figures and get

$$\text{Load per hour, Btu} \times 0.0136 = \text{cfm of air}$$

We can tabulate the air requirements for the sample building as shown in Table 8.13.

TABLE 8.13

Room	Load, Btu/hr	Cfm	Add 25%, cfm	Design, cfm
A	20,000	271	68	339
B	22,000	300	75	375
C	8,000	109	27	136
D	9,000	123	31	154
E	8,000	109	27	136
Total............	67,000	...	..	1,140

Sizing the Ducts. The diagram of Fig. 8.11 shows us that a horizontal duct emerging from the heater must carry the entire load as far as the branch to room *E*. To do this, assuming a main duct velocity of 800 fpm, we shall need to provide 1,140/800 = 1.425 sq ft of cross-sectional area.

In terms of square inches, this means 1.425 × 144, or 205.2. Whether we select a round duct or a rectangular duct is a matter of preference; generally a rectangular duct makes a better appearance, and branches can be faired into it with less difficulty. On the other hand, if circular ducts are used, some reduction in size can be expected, because friction is less in the circular duct. Friction decreases in the rectangular duct as its form approaches a square, and this fact may be used as a point of departure, when dimensions are being decided.

A square duct of 205 sq in. cross section would be approximately 14.3 in. on a side, but perhaps our headroom in the basement is limited, and we cannot afford to give 14.3 in. to a duct. If we can spare 10 in., then a duct 10 by 20.5 in. will do very well. Let us adopt a conventional 10- by 20-in. duct, which is close enough.

The next section of the main duct, after the branch to room *E* has been taken off, must handle 1,140 cfm − 136 cfm = 1,004 cfm. For this volume at 800 fpm, we need approximately 1.255 sq ft of cross-sectional area. Converting to square inches, we have a little less than 181, and a duct 18 by 10 in. will serve nicely. Therefore we mark down these dimensions on the diagram.

Now we come to a point of division, with 714 cfm making a right-angle turn to the second-floor rooms, and 290 cfm continuing straight ahead to rooms *C* and *D*. We shall consider the larger load to be moving through the main duct, and on the basis of this assumption $^{714}\!/_{800} = 0.9$ sq ft approximately, or 130 sq in. Still holding to our 10 in. of depth, we have a 13- by 10-in. duct to mark on the diagram. These dimensions will hold up to the room *A* branch.

For the branch to rooms *C* and *D*, we shall use a branch velocity of 700 fpm, which calls for a duct 0.414 sq ft in area, or 60 sq in. Here we can drop the depth to 6 in., and spread out horizontally 10 in.

Now let us look at the individual room connections. To room *A*, we have 339 cfm and will supply it at a velocity of 600 fpm. This produces a duct approximately 10 in. wide and 8 in. deep, very comfortable dimensions since the form of the duct is approaching the square, and we are at some distance from the heater. Room *B* will take a 10- by 9-in. duct, or a 12- by 8-in. duct, the horizontal dimension being the first one given. Incidentally we should note that shop and design practice call for the horizontal dimension to be given first on the drawings, thus avoiding confusion.

Rooms *C* and *E* call for 136 cfm each, and at 600 fpm, the final runs of duct in the basement to these rooms will be 6 by 5 in. or 8 by 4 in., respectively.

Room *D* supply is 154 cfm, and the duct will be 8 by 5 in.

Now examine the stack sizes. If we allow 600 fpm in the stack to room *A*, the 339 cu ft will require 80 sq in. of cross section.

For a stack only 3¼ in. deep, this means we must provide approximately 25 in. of width. However, the clear space between wall studs is normally only 14 in. What shall we do?

We have three possible plans from which to choose:

1. We can frame the studding around the stack to yield the 25 in. that is needed. This is the plan generally followed. In most cases no difficulty is encountered in carrying it out.

2. We can investigate the possibility of increasing the partition depth to 6 in., thus enabling us to squeeze a 5¼-in. stack between studs on a standard spacing. Architects do not like to do this as a rule, since it tends to upset the proportions of a room, and it certainly wastes space in those sections of the 6-in. partition that do not contain the stack.

3. We can divide the load and use two stacks instead of one. Often this plan is the best of all, for a room that poses a load too large for a single normal stack certainly enjoys more uniform heating if the warm air is introduced at several points.

We should keep in mind always that the prime purpose of a heating system is to heat. Consequently the delivery of the proper quantity of warm air, to the suitable areas of a room, is our basic responsibility. Whether we discharge this responsibility by means of one stack or six is of secondary importance.

All the wall stacks in our subject building will be sized similarly. Generally, we will design for a velocity of about 500 fpm, for these stacks are a far departure in form from the ideal square of minimum friction loss; 600

fpm, however, will do if necessary. Dangers from high velocity are two: noise and excessive friction loss. We shall avoid both, as we can.

We now have all the data for our diagram (Fig. 8.11) and proceed with the finished drawing of ducts and stacks for this building. What we put down now will reach the contractor eventually on a blueprint, and that is the way he will make the installation.

Duct Materials and Design. The best material for an air-carrying duct appears to be smooth metal. This seems to be an obvious conclusion, but the plentiful existence of rough wood ducts (as between joists) or ducts of brick, stone, and concrete suggests that we do not always do what we know to be best.

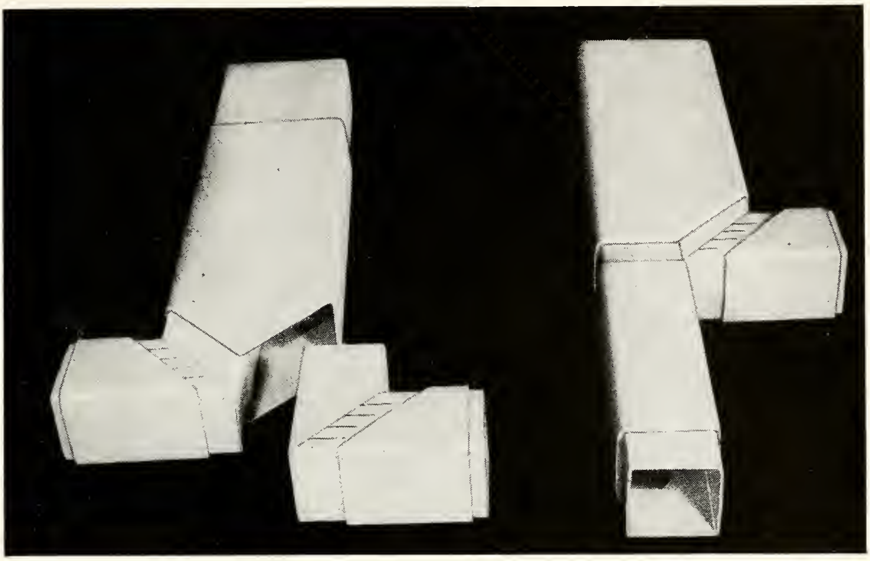


Fig. 8.12. Asbestos duct and fittings. Turning vanes may be inserted in the elbow slots if desired. *(Courtesy of The Philip Carey Manufacturing Company.)*

One material that is receiving substantial acceptance is preformed asbestos. Ducts of this smooth asbestos show approximately the same friction values as metal, provide their own insulation, and effectually dampen noise, thus permitting somewhat higher velocities. Figure 8.12 illustrates a straight length of this preformed duct, together with a 90-deg elbow. Turning vanes are mounted in the elbow to achieve a smooth change of direction in the air flow.

Table 8.14 sets up some standard dimensions for ducts of this material.

The friction loss in straight sections of round duct can be taken from Fig. 8.13. To convert this into rectangular duct, we must refer to Table

8.15 and select two sides that together will provide a cross-sectional area equivalent to that of the round duct.

For example, suppose that we use asbestos duct for the main run in Fig. 8.11 and take advantage of the sound-deadening faculty of the asbestos by stepping up the velocity to 1,000 fpm. At this rate our 1,340 cfm will require 193 sq in. of duct cross section, equivalent approximately to a round duct 16 in. in diameter. By consulting Table 8.15, we observe that a rectangular duct 12 × 18 in. is about right for equivalent friction.

TABLE 8.14. ASBESTOS DUCT DIMENSIONS, INCHES*

2½ × 5½	4 × 5½	8½ × 8½	14½ × 14½	17½ × 17½
2½ × 8½	4 × 8½	8½ × 11½	14½ × 17½	17½ × 23½
2½ × 11½	4 × 11½	8½ × 14½	14½ × 20½	20½ × 20½
2½ × 14½	4 × 14½	8½ × 17½	14½ × 23½	20½ × 23½
2½ × 17½	4 × 17½	8½ × 20½		23½ × 23½

* Inside. Length of each section, 3 ft 0 in. Wall thickness, for winter heating and air conditioning, ½ in. Wall thickness, for summer air conditioning, 1 in.

TABLE 8.15. DIMENSIONS OF RECTANGULAR AND ROUND DUCTS OF APPROXIMATELY EQUAL FRICTION*

Rectangular Dimensions, In.	*Round Diameter, In.*	*Rectangular Dimensions, In.*	*Round Diameter, In.*
4 × 8	6	9 × 12	12
4 × 10	7	9 × 14	12
5 × 8	7	9 × 16	13
5 × 10	8	9 × 18	14
5 × 12	8	9 × 20	15
5 × 14	9	10 × 12	12
6 × 8	8	10 × 14	13
6 × 10	8	10 × 16	14
6 × 12	9	10 × 18	15
6 × 14	10	10 × 20	16
7 × 10	9	12 × 14	14
7 × 12	10	12 × 16	15
7 × 14	11	12 × 18	16
7 × 16	12	12 × 20	17
8 × 10	10	12 × 22	18
8 × 12	11	14 × 18	17
8 × 14	12	14 × 20	18
8 × 16	12	14 × 22	19
8 × 18	13	14 × 24	20

* *Note:* These data have been abstracted with permission from the ASHVE Guide, 1948 Edition. The diameters of the round pipes have been rounded off, to avoid fractional sizes. This is a departure from the original appearance of the equivalents, as published in the Guide.

According to Fig. 8.13, a velocity of 1,000 fpm in either of these ducts will develop only 0.09 in. friction loss per 100 ft of duct. Elbows will add friction substantially as shown in Table 8.16.

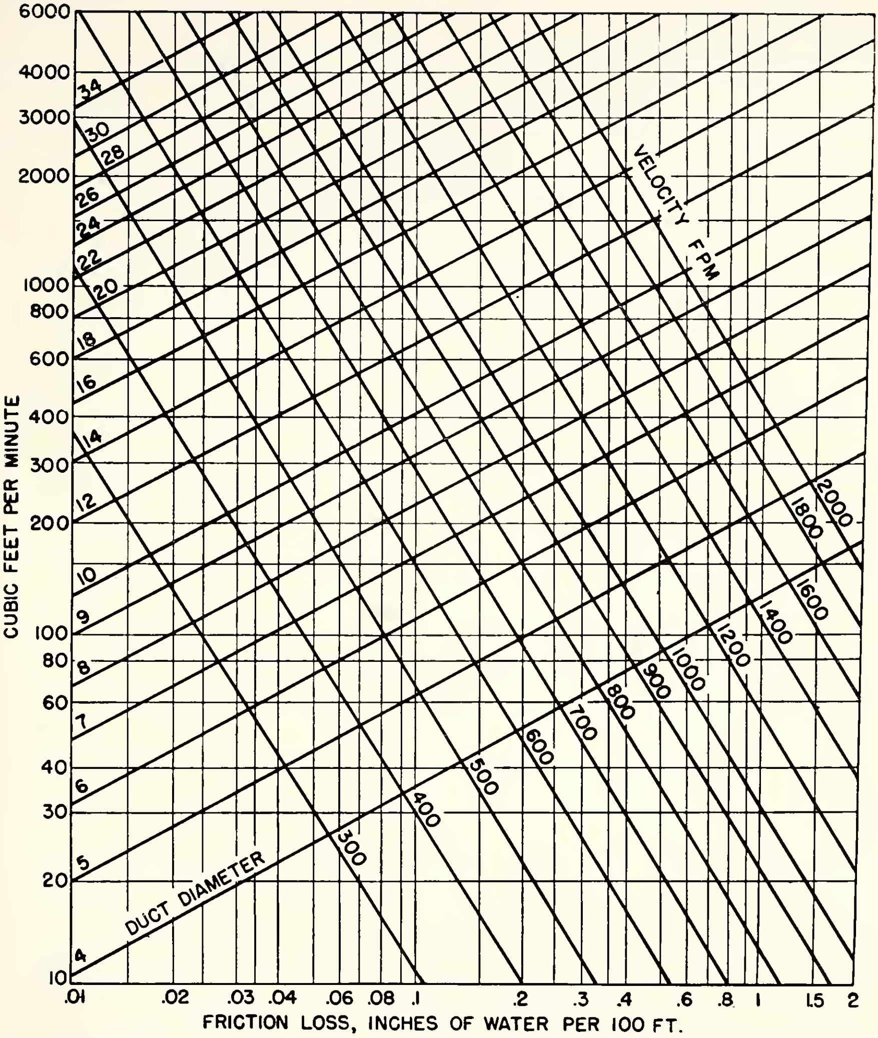


Fig. 8.13. Friction loss for air in straight round ducts. (*Abstracted with permission from 1948 ASHVE Guide, Chap. 4.*)

Galvanized Iron. Here is the old stand-by for duct construction. A galvanized sheet is thicker than the base metal by the film thickness of the galvanizing, usually about 0.004 in.

Our concern is mainly with designing our duct for the proper thickness of metal, together with a selection of the most suitable joint form and the need for bracing, if any. As usual, somebody already has provided for us a dependable basis on which to make our decisions. In this instance, our

TABLE 8.16. FEET OF STRAIGHT ASBESTOS DUCT EQUIVALENT TO ELBOW RESISTANCE*

Small dimension of duct, in.	Velocity	
	800 fpm	1,000 fpm
2½	4 – 7	4 – 7.4
4	5 – 9.1	5 – 9.7
5½	5.9–11.6	6 –12
8½	8.3–15.2	9.2–15.6
11½	12.2–16.7	12.0–18.2
14½	15.4–20.0	16.3–20.5
17½	18.5–21.8	19.5–23.0
20½	21.9–23.8	23.3–24.4
23½	26.3	26.9

* *Note:* In each case, the *minimum* loss is for a square duct, with all sides equal to the small dimension given. The *maximum* loss is for the small dimension shown, with 23½ in. for the long side. For exact values of the intermediate dimensions, tables should be obtained from the duct manufacturer. Manufacturer's tables will show losses for various velocities up to 2,400 fpm.

good friend is the American Society of Heating and Ventilating Engineers which has compiled the recommendations listed in Table 8.17.

TABLE 8.17. STANDARDS FOR DUCT CONSTRUCTION

U.S. Standard gauge	Max side, in.	Type of joint	Bracing
26	Up to 12	S, drive, pocket or bar slips on 7′10″ centers	None
24	13–24	Same	None
24	25–30	Same	1″ × 1″ 1 × ⅛″ angles, 4′0″ from joint
22	31–40	Drive 1″ pocket or bar slip, on 7′10″ centers	Same
22	41–60	1½″ angle connections, or 1½″ pocket or bar slip with 1⅜″ × ⅛″ bar reinforcing on 7′10″ centers	1½″ × 1½″ × ⅛″ angles 4′0″ from joint
20	61–90	Same as above, but with reinforcing on 3′9″ centers	Same as above but only 2′0″ from joint

The various joints referred to in Table 8.17 are illustrated in Fig. 8.14. Sometimes large flat duct surfaces tend to breathe and vibrate. To check this phenomenon, we cross-break the metal, which means to impress a stiffening crease across the surface. The cross-breaking treatment should be applied to all ducts 18 in. and larger, if these ducts are not insulated. In lieu of cross breaking, the same result can be accomplished by using metal two gauges heavier than is indicated in Table 8.17.

Aluminum. Aluminum ducts achieve a notably handsome appearance. They tend to cost a little more than galvanized ducts, but the relative prices of the two materials fluctuate, depending on the current demand and availability.

The design recommendations for duct construction which appear in Table 8.17 for galvanized ducts apply to aluminum ducts as well, provided we make a single correction. This correction concerns the difference in gauges that obtains between galvanized and aluminum sheets.

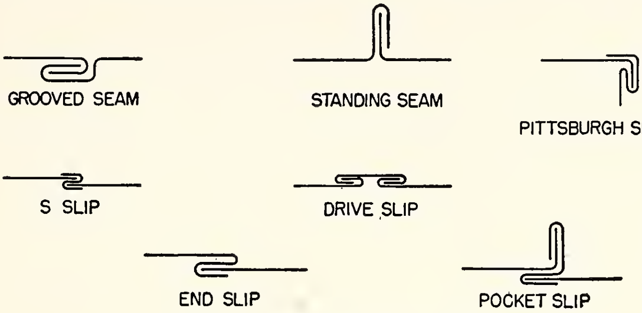


Fig. 8.14. Types of sheet-metal joints.

Aluminum is gauged according to the American or Brown & Sharpe standards, and compared with the U.S. Standard gauge of galvanized sheets, the differences appear as follows (Table 8.18):

TABLE 8.18. COMPARATIVE GAUGES OF ALUMINUM AND IRON*

Gauge No.	Thickness, in.		
	Aluminum	Steel	Iron
26	0.01594	0.0184	0.0188
24	0.02010	0.0245	0.0250
22	0.02535	0.0306	0.0313
20	0.03196	0.0368	0.0375
18	0.04030	0.0490	0.0500
16	0.05082	0.0613	0.0625

* For galvanized thickness, add 0.004 to the steel and iron cores shown in this table.

In consequence of these differences, we cannot use the B & S aluminum gauge interchangeably with the U.S. Standard gauge, but must consider metal thicknesses, in inches. As an example of how this is done, consider a duct with a maximum side of, say, 26 in. According to Table 8.17 the duct must be fabricated of No. 24 gauge galvanized iron or steel. The steel thickness of 24 gauge is 0.0245 in., and an aluminum sheet at least equal to this in thickness is No. 22 on the B & S gauge, which has a thickness of 0.02535 in. Thus, if we build our duct of aluminum, it will be of No. 22 gauge aluminum and nothing lighter.

Aluminum can be riveted, welded, soldered, and brazed. Soldering and brazing offer some difficulties, and care must be exercised to obtain fluxes specifically compounded for aluminum.

Aluminum ducts may be painted with any good grade of oil paint, but the first coat or "undercoater" must be designated for aluminum. In any event, the surface should be thoroughly cleaned before application of the paint. A solvent such as naphtha does an excellent job and may be used in washing down the surface by hand.

Friction in Metal Ducts. In order to size the air-moving fan in a mechanical heating system, we must arrive at some valuation of the resistance in the system.

The chart (Fig. 8.13), friction of air in straight round ducts, augmented by Fig. 8.15, which shows pressure loss in elbows, enables us to calculate duct losses in any system.

Let us determine the pressure loss in the ducts that distribute heat to the house in Fig. 8.10. Our first step is to identify the longest equivalent run, which seems to be from the heater to the register in room *B*. Next, we set down the sections in the run as shown in Table 8.19.

TABLE 8.19

Section	Straight length, ft	Equiv. elbow length	Total equiv. length	Cfm	Velocity, fpm	Friction loss
From heater to branch to room *E*	4	None	4	1,340	800	$\frac{0.05 \times 4}{100}$
From branch *E* to branch supplying rooms *C* and *D*	7	1	17	1,004	800	$\frac{0.067 \times 17}{100}$
To branch to room *A*	10	None	10	714	800	$\frac{0.08 \times 10}{100}$
From branch to room *A* to register, room *B*	12	1	22	375	600	$\frac{0.06 \times 22}{100}$

The fractions in the column headed Friction Loss are necessary because the losses shown in Fig. 8.13 are for 100-ft sections. Solving the fractions, we get $0.002 + 0.012 + 0.008 + 0.013 = 0.035$ in. W.G.

The elbow equivalent chart (Fig. 8.15) permits us to evaluate projected designs from the view of duct radius. For the purposes of the pressure losses just computed, we have used the square elbow on the basis of an elbow radius equal to 100 per cent of the duct width. In practice, we should use the greatest possible radius and seldom less than 75 per cent of the duct width.

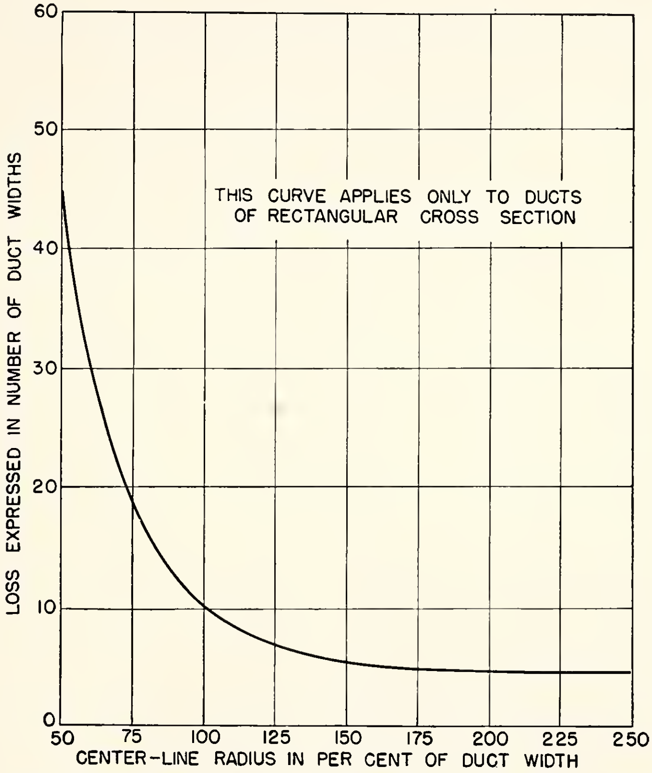


Fig. 8.15. Friction loss in duct elbows. *(This curve has been drawn from data appearing in the 1948 ASHVE Guide.)*

Registers. Mechanical circulation systems allow us to use our discharge registers to the very best advantage. In Fig. 8.16, we see how well the direction of air discharge can be controlled. This means we can place a register near a wall if necessary, and yet spread comfort, fanlike, to those parts of the room where it will do the most good.

Figure 8.17 illustrates the arrangement of the shutters and louvers in a register, which together enable us to accomplish such pleasing distribution of the discharge.

Selection of a register involves consideration of the throw. We have already discussed this factor of throw in step (7) of the design procedure. Now let us take a specific case and see how it works.

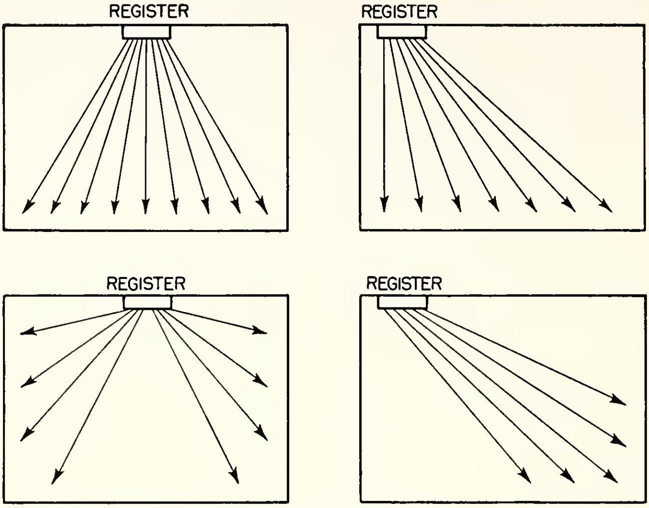


Fig. 8.16. Possible forms of air discharge from registers.

For room *D* in Fig. 8.11, we must deliver 154 cfm from a high wall register and project this discharge across the room to the opposite wall. Now we look at the curves on Fig. 8.18, find our 154 ft about midway between the 150- and the 160-cfm curves, and run down the curve to the intersection with the vertical 11-ft throw line. Apparently we can secure the desired throw by selecting either a 14- by 4-in. register or a 12-by 5-in. register.

Fig. 8.17. Control shutters of a modern register. (*Courtesy of Hart and Cooley Manufacturing Company.*)

The chart we have used for this demonstration (Fig. 8.18) is suitable for one manufacturer's registers only. If we buy registers of another brand, we shall need a different throw chart. This fact must be remembered when an actual job is undertaken; neglect can contribute to an unsatisfactory installation.

The same situation must be considered for the selection of baseboard registers. Throw is not important here, but diffusion is. Each manu-

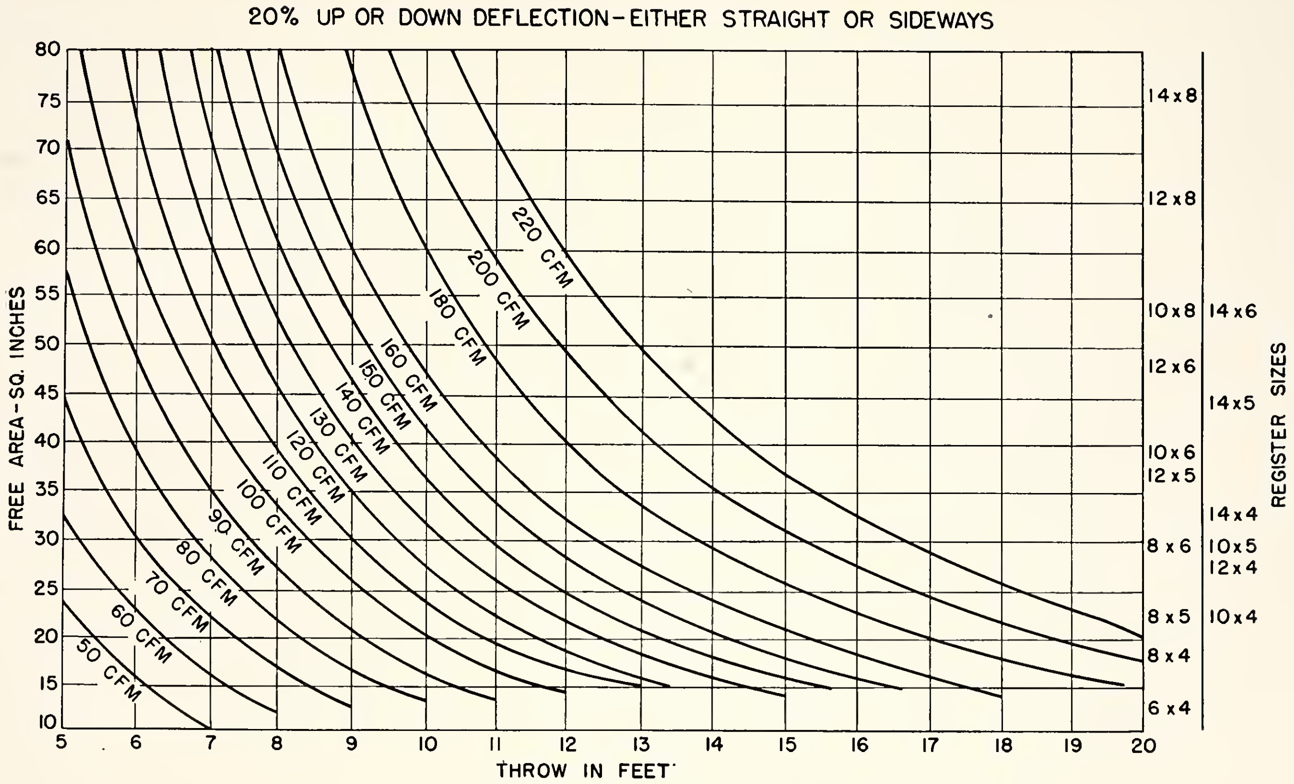


Fig. 8.18. Typical throw curves for registers.

facturer will report on the capacity of his baseboard registers. From some registers, the air emerges in a level plane; others permit a downward deflection of 20 deg; for all we may have sideway deflections if we want them.

The use of high wall registers or baseboard registers is a matter to be decided in each case. Some people will have only high wall registers in their homes; others insist on baseboard units.

The cost of construction can have an influence on our decision. Short runs to the registers are desirable, and if the warm air is supplied from attic ducts, with stacks descending in the partitions, then high wall registers will require the shortest stacks. If the air is conveyed upward from the basement, baseboard outlets become the shortest route between the heater and the room.

Return Arrangements for a Mechanical Warm Air System. Many systems will be observed that employ a return from each room. This elaboration seldom is necessary. The ASHVE has run tests that depreciate the importance of location where a return inlet is concerned. Obviously, the coldest air in the room will find its way to the intake, an action that is as natural as the running downhill of water. Gravity is the motive force in both cases.

One or two return inlets will serve most installations with adequacy. The rooms are placed under a tiny positive pressure by the warm air being forced into them by the fan. To make space for this influx of new and heated air, the colder air at floor level falls down the intake stack if present, otherwise oozes out underneath the door if the room door is closed, or eddies through door-panel louvers if such openings have been provided for its exit.

Just see that a route is provided for air movement from every heated space to the return inlet, and nature will do the rest. Sometimes, a half inch planed off the bottom of a room door is sufficient, with a return inlet grille 20 ft down the hall.

When sizing return ducts for a mechanical circulation system, we should consider the lower temperature and reduced volume of the returning zephyrs. For example, a pound of air entering a room at 150 F occupies approximately 15.4 cu ft; but the same pound leaving at only 70 F has shrunk to 13.4 cu ft.

Now let us see what kind of returns we shall need for the building in Figs. 8.1 and 8.10.

For the second floor supply we have a total of 714 cu ft of 150 F air. Since $13.4/15.4 = 0.87$, our return will amount to only 87 per cent of 714, or 621 cu ft.

If we permit a velocity of 600 fpm in the returns, we must provide a cross-sectional area of 621/600 = 1.04 sq ft, or 150 sq in. With a partition yielding, say, 3.5 in. in depth for the stack, our metal-sided air chute will be 43 in. wide.

Enclosing so wide a stack introduces problems of wall framing, as we observed in our considerations of the supply stacks. It can be done, but any duct or stack that is designed with the long side more than 10 times the length of the short side is definitely a bad design. Friction loss increases tremendously.

The best thing to do is illustrated by the expedient of setting the duct inside a closet, with the intake grille mounted in the closet wall. No space is taken from the room's floor area, and the necessary height of the stack inside the closet (just enough to accommodate the height of the grille) usually is insufficient to conflict with the hanging of ordinary clothing. A stack 10 by 15 in., for instance, which provides enough area for the building in Fig. 8.10, is easily installed in a closet.

Ducts and stacks are seldom items of beauty, and in most of our design work, the less we see of them as they go about their daily work, the better.

The other stack in Fig. 8.10 is sized in precisely the same way. With only 371 cfm to handle (87 per cent of 426 cfm), we shall need 90 sq in. of cross-sectional area. This will give us a stack 26 in. wide, approximately, and we can set it easily in the wall by skipping one vertical stud.

In the basement, we can step up the velocity to 800 fpm. At this velocity, a 12- by 8-in. duct will do very well for the second-floor return, and a 10- by 7-in. duct for the first floor.

We need not worry about pitching the return ducts for a mechanical circulation system, since all air is moved by a fan. Nevertheless, a slight slope toward the heater will make the flow easier in a long, generally horizontal duct. Provide it as a luxury if you wish, not a necessity.

Standard Sizes for Ducts, Pipes, and Fittings. The National Bureau of Standards of the U.S. Department of Commerce has adopted certain standard sizes and shapes for all the component sheet-metal parts that are used to form a warm air heating system. These standards are printed in a pamphlet "Simplified Practice Recommendation R207-45, for Pipes, Ducts, and Fittings for Warm Air Heating and Air Conditioning," copies of which may be obtained from the Government Printing Office, Washington, D.C.

The objective of the standardization is to permit prefabrication of the parts, without imposing on the fabricator an excessive inventory.

Some installation contractors may prefer to buy prefabricated piping and ducts, and in designing for this group, the standards should be used. Other contractors operate their own shops, and every installation is considered a tailor-made job; then we can size the ducts to suit ourselves more or less, within the limitations of the job.

Insulation for Ducts. All warm air ducts should be insulated as specified in the sections dealing with insulation for gravity systems. One important variation, however, should be noted, where mechanical circulation is concerned. Return air ducts, if traversing spaces where the ambient temperature of the air is cooler by more than 15 deg than the returning air, may be insulated in many cases with justification. Heat lost in the return duct must be made up in the heater; every lost Btu is replaced by Btu from the fuel. In short, bare return ducts can be an expense.

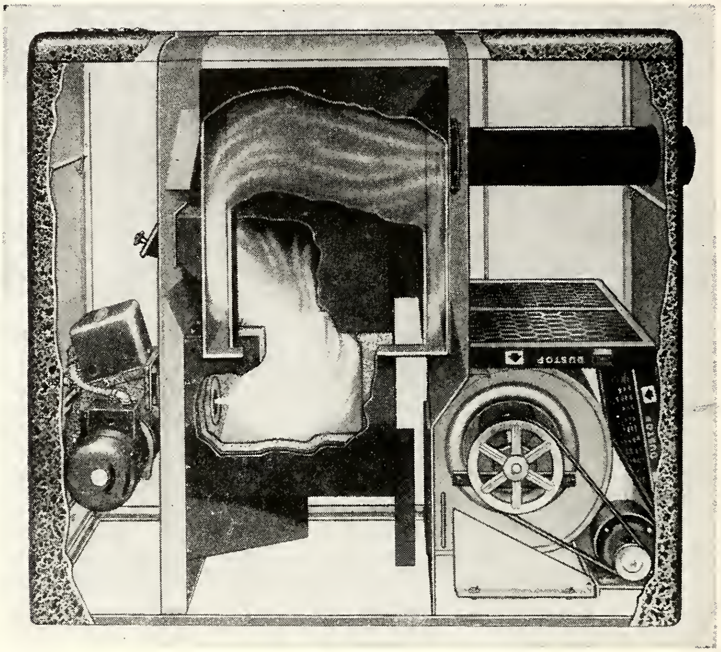


Fig. 8.19. Winter air conditioner incorporating fan, humidifier, filters, and oil burner. (*Courtesy of Dowagiac Steel Furnace Company.*)

Mechanical Circulation Heaters. Figure 8.19 illustrates the arrangement of a conventional mechanical circulation heater.

Manufacturers call these units "winter air conditioners." The term is based on the following services performed within the heater:

1. The air is filtered by a dry-mat barrier that is readily replaced at the season's end, or before if it clogs up with dirt.

2. The air is humidified by the simple procedure of setting a pan of

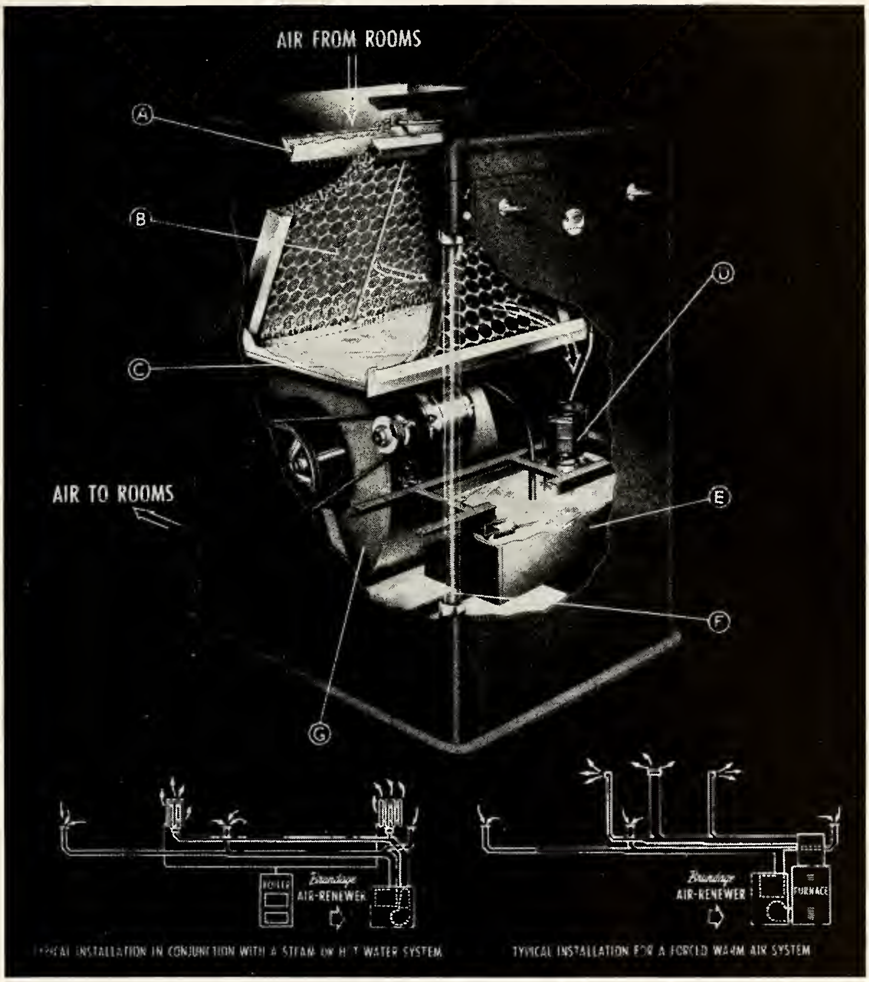

AIR FROM ROOMS
A
B
C
D
E
AIR TO ROOMS
F
G
BOILER
Brundage
AIR-RENEWER
TYPICAL INSTALLATION IN CONJUNCTION WITH A STEAM OR HOT WATER SYSTEM
Brundage
AIR-RENEWER
FURNACE
TYPICAL INSTALLATION FOR A FORCED WARM AIR SYSTEM

Fig. 8.20. A form of air treatment unit that attacks cold germs by a glycol spray and violet-ray lamp. *A.* Perforated channel to distribute liquid over filter. *B.* Filter impregnated with glycol-water solution. Supported by stainless steel racks. C. Stainless steel inner casing. *D.* Pump and float valve assembly. *E.* Glycol reservoir. *F.* Ultraviolet germicidal lamp. *G.* Blower for circulating air. (*Courtesy of Brundage Company.*)

water in the air stream. Water in the pan is maintained at the proper level by an automatic float valve.

3. The temperature of the air issuing from the registers is under close control at all times. A delayed action switch prevents the fan from operating if the air in the heater bonnet is less than, say, 140 F. Another

switch instantly terminates combustion for oil or gas, and dampens combustion for stoker-fired units, if the warm air temperature rises to say 175 F.

The "winter air conditioner" does not kill germs, however, or destroy air-borne viruses in its customary form. In Fig. 8.20, we see a unit that is designed to do all these things. A glycol spray is introduced into the air stream where it promptly vaporizes, and the result, according to reports, is fewer colds for all persons served by the system.

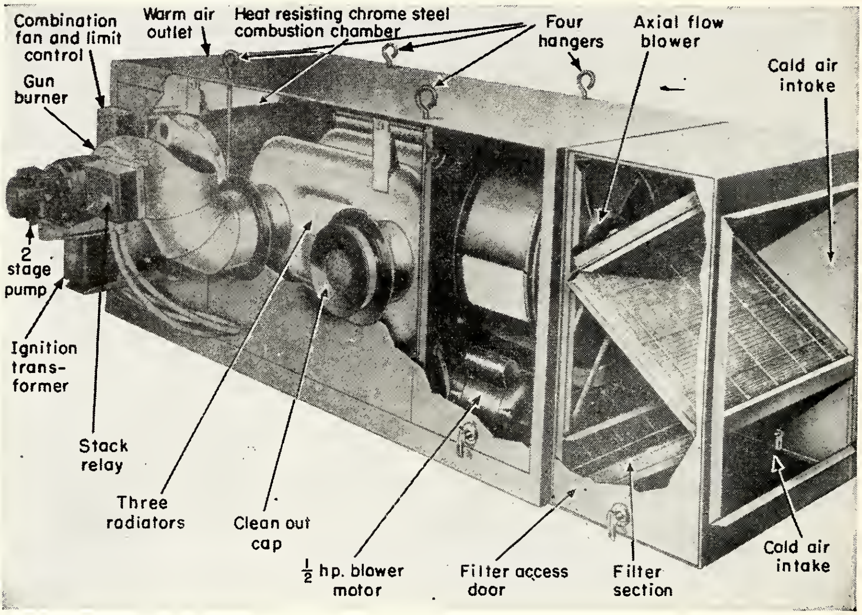


Fig. 8.21. The Superfex suspension heater. (*Courtesy of Perfection Stove Company.*)

This particular cabinet also contains violet-ray lamps. Ultraviolet rays are not welcomed by many forms of bacteria; the result is usually death for the organisms after even short exposures. Reported tests indicate that over 95 per cent of disease-causing germs are destroyed by the deadly light as they pass through the machine's illuminated chamber.

Figure 8.21 illustrates a recent type of heater, designed primarily for shops and factories, and intended to be hung from overhead supports. However, a long, low shape, such as this, is suited to many applications. For example, the currently favored one-story homes without basements can be heated admirably from units such as these, simply by setting them on the ceiling joists under the low roofs. Four feet of headroom enables us to have a heating plant in a space that is little suited to anything else.

The ducts then run across the joists to drop stacks to high wall registers. The single return is taken from some convenient spot at the ground-floor level.

To keep down noise, it is wise to set these units on a mat of cork 3 in. thick. Nevertheless, they operate with surprising silence; we must listen very closely to hear them run.

Fire departments regard these attic installations with much favor. A fire originating in the heating plant goes straight up through the roof, and building damage is minimized.

To size the winter air conditioner, we shall follow the procedures outlined previously in this chapter. That is, we add a percentage to the calculated load for pickup and intermittent operation, note how many cubic feet of air per minute we need, and then refer to a manufacturer's catalogue and put a check mark on the unit that equals, or next exceeds, our requirements.

In the building shown in Fig. 8.10, our calculated heat loss is 67,000 Btu per hr. If we add 50 per cent to this, we shall require a unit that will deliver 100,500 Btu per hr. As for the fan, 1,140 cfm *at the registers* of 150 F air will be quite satisfactory.

We should note that the fan handles cool air, on the return side of the heater. Obviously, a smaller volume is involved at this point in the system; consequently the manufacturer's catalogue can mislead us if we do not think carefully. As we observed before, our 150 F air shrinks to 87 per cent of itself if cooled to 70 F, and to achieve 1,140 cfm at the registers, 992 cfm entering the fan is enough.

Total Friction Loss in a System. The delivery of a fan is affected by the pressure against which it must work; thus we are interested in the total friction loss of the system. For the example in hand, we have already calculated a supply side loss in the ducts of 0.035 in. W.G. For the two return ducts, we will assume an equal loss, although it seems likely to be a little smaller. Some figures and assumptions are given in Table 8.20.

TABLE 8.20. FRICTION LOSS IN A SYSTEM

Location of Loss	*Amount, In. W.G.*
Supply ducts	0.035
Return ducts	0.035
Register (from manufacturer's catalogue)	0.030
Return grille (from manufacturer's catalogue)	0.060
Filters (from manufacturer) assumed	0.030
Heater (from manufacturer) assumed	0.050
Total	0.240

On the basis of the above summary, our fan must do what we want it to do despite a system resistance to air movement of 0.24 in. W.G. Since most of the units offered commercially are designed to meet a resistance of from 0.25 to 0.375 in. W.G., our system is readily accommodated. Occasionally we will have a condition that imposes resistances in excess of the standard fan capacity. This situation can be met in one of two ways: either buy a larger heater or change the fan. The latter choice is usually preferred, as being the cheaper.

The Complete Design. We have taken the building in Fig. 8.10 as an example and for its requirements have sized the ducts, stacks, registers, grilles, and heater. We have determined the size and capacity of the fan. But our work is not yet completed. There still remains the need to make the drawings and to write the specification. The art of mechanical drawing is not within the scope of this book. As for the preparation of a specification, the fundamental precept is to state what we want as clearly as we can.

A discussion of specifications, including the items to be included, appears in Chap. 21.

To carry through any design job to completion, we must do the following:

1. Make all essential calculations.
2. Make all essential drawings.
3. Write a specification that, in conjunction with the drawings, informs contractors of exactly what is wanted. When we achieve this point in our labors, design work is ended.

Variants of the Mechanical Systems

Mechanical warm air systems provide a rich field for new ideas. Manufacturers are originating and developing new arrangements and new plans of heating, to increase the sales of their equipment. Engineers also are experimenting with shapes, sizes, and the governing of air flow through ducts. Description of some interesting developments follows.

Room Induction System. In this design, ducts are round and small, only 3½ in. in diameter. They carry high-temperature air, around 195 F, at a velocity of about 1,100 fpm to special mixing cabinets set in or against the wall of a room. The velocity of the air entering the cabinet inducts room air from near floor levels, mixes this room air with its own high-temperature bulk, and discharges the blend from a higher register, usually at about 135 F.

Return grilles and ducts are conventional in design, but are smaller, naturally, since only a part of the heating air actually circulates through the heater.

Savings in installation costs are claimed for this design, based on the comparative smallness of the ducts. Fan horsepower is not increased over the conventional design, because the volume handled is less, and the friction loss from high velocity is kept within bounds for the same reason.

Control of temperature and register discharge is achieved by means of an adjustable nozzle valve on the hot air supply, set within the mixing cabinet.

In houses without basements, the high-temperature pipes are run through the attic, and the movement of air in the mixing cabinet is reversed, so that the room air enters by way of the upper grille and is discharged at baseboard level from a register. Figures 8.22 and 8.23 illustrate the arrangement of this system.

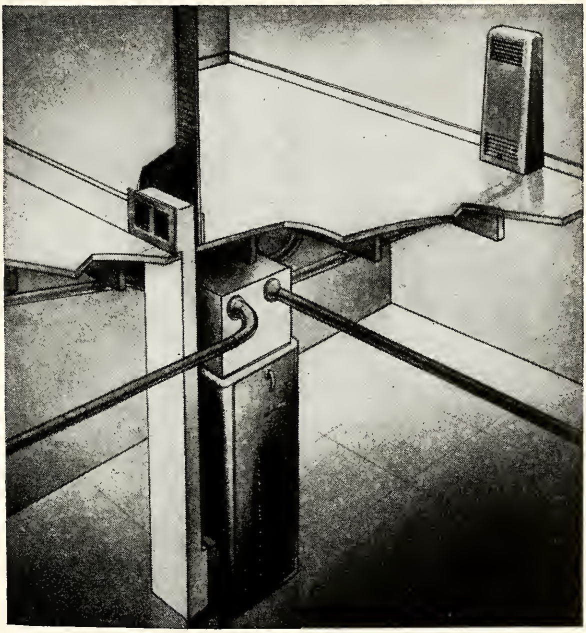


Fig. 8.22. The "Blend-Air" high velocity system arranged for distribution from a basement heater. *(Courtesy of the Coleman Company, Inc.)*

At present, equipment is available only for comparatively small heating loads, with 100,000 Btu per hr as a maximum.

Air Wall System. Figure 8.24 shows a typical register for this arrangement of forced circulation heating. As with the room induction scheme, small pipes, 4 in. in diameter, are used to convey the warm air from the heating unit to the discharge register.

On arrival at the register, the high velocity air is slowed down in a sort of plenum box behind the grille, and then emerges in a vertical fan that rises and opens across the surface of the wall.

Design procedures locate these registers in the outside wall of a room, contrary to conventional warm air practice. The cold wall is thus heated, and affords some measure of radiant heating while the air itself eventually diffuses across the ceiling. Returns are taken from inside walls through similar round ducts.

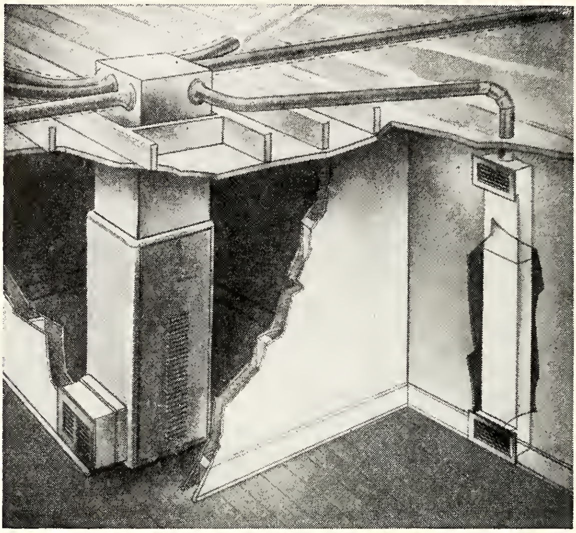


Fig. 8.23. High velocity system with attic ducts. (*Courtesy of the Coleman Company, Inc.*)

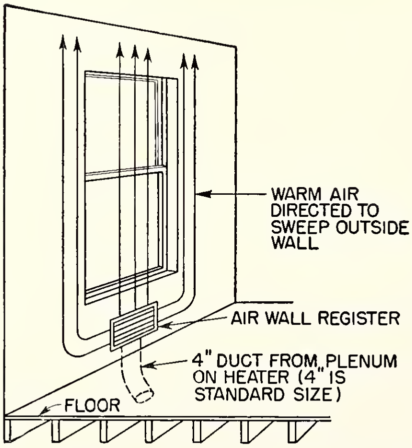

WARM AIR DIRECTED TO SWEEP OUTSIDE WALL
AIR WALL REGISTER
4" DUCT FROM PLENUM ON HEATER (4" IS STANDARD SIZE)
FLOOR

Fig. 8.24. The General Electric high velocity system.

The air wall system claims installation savings, and a high measure of comfort. Nothing is reported as yet on wall discoloration, if any. We may need several years of experience to determine the true importance of this factor.

Equipment for this system is available in somewhat larger sizes. Gas fired heaters may be had for loads up to 168,000 Btu per hr; with oil-fired units, 155,000 Btu per hr.

Any Warm Air System. Figure 8.25 shows a smoke maker that is extremely useful to the heating engineer for checking drafts and air movements in any room.

Figure 8.26 illustrates a recently developed thermostatic register that promises much assistance in providing different room temperatures from a common system. Each room thus comes under the temperature control of its occupants.

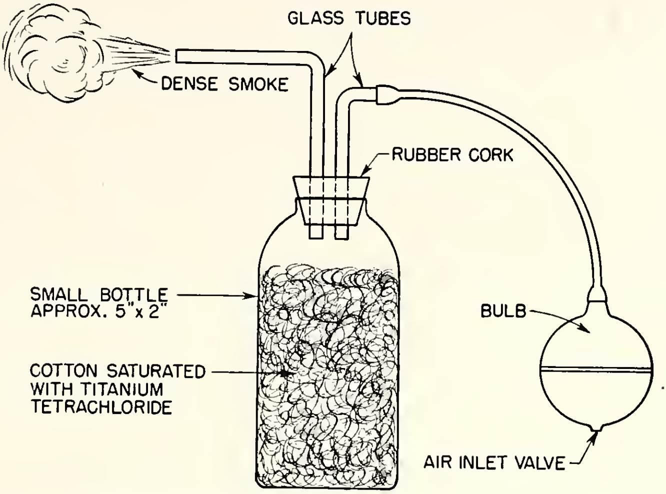


Fig. 8.25. Smoke maker for showing air movement.

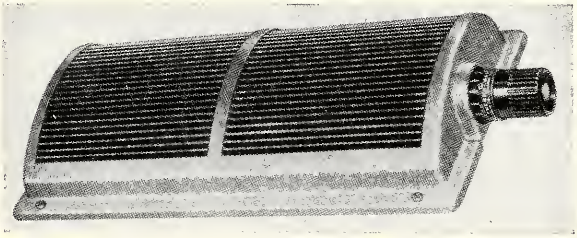


Fig. 8.26. Dole thermostatic register for warm air systems.

Questions

1. List the advantages and disadvantages of a gravity warm air heating system.

2. Why do long horizontal ducts sometimes cause trouble in delivering warm air?

3. Do you locate warm air registers high or low in a wall? Why?

4. Name at least three absolutely essential considerations in designing a gravity system.

5. What is the motive power of a gravity system? How does it operate?

6. Lay out a gravity system for your own house, sizing the leaders, ducts, registers, and stacks as recommended in this chapter.

7. What are the advantages of a mechanical circulation system compared with a gravity system?

8. Enumerate the eight cardinal steps to follow in the procedure of designing a forced circulation system.

9. How much excess capacity will you provide for a system in your neighborhood, expressed as a percentage of actual load calculations?

10. List the velocities that will furnish satisfactory noise and friction levels for (*a*) main ducts, (*b*) branches, and (*c*) stacks.

11. What is throw as it refers to a register? Why is it important?

12. What would you do if your stack calculations produced a stack too large to be placed within the normal space between studs? Why?

13. Lay out a forced circulation warm air system for your own home, sizing all component parts.

14. Where will you use insulation, and why?

15. Will you select high wall or baseboard registers for your own house? Why?

16. Calculate the friction loss in your own system. Against what resistance in inches of water must the fan deliver its necessary volume?

17. What is the important point to consider when selecting aluminum for ducts?

18. Do you prefer aluminum or galvanized sheets? Why?

19. How do you feel about multiple return inlets compared with only one or two? Why do you prefer one or the other?

20. Describe two high-temperature high velocity systems.

CHAPTER 9

PANELS, BASEBOARDS, AND UNIT HEATERS

The Panel Picture

How Panel Heating Works. Every exposed cast-iron radiator is, in effect, a form of panel heating apparatus. There is always an interchange of heat between surfaces of unequal temperature, this interchange occurring in the form of straight-traveling, invisible heat rays, known as radiation. These rays actually constitute a form of energy moving at the astounding speed of light, 186,000 miles per second. When they strike materials capable of absorbing heat, such as our bodies, the impact converts the energy into heat, and the materials are warmed accordingly.

Since these rays move in severely straight lines, our conventional radiator is prevented by its inherent design from dispatching many of them usefully: rays are absorbed by neighboring and opposed sections of the same radiator almost at the moment of their births. The conventional radiator therefore accomplishes its heating destiny mainly by convection.

Panel arrangements are planned to reverse the proportions of radiation and convection emissions of heat as observed in the familiar radiator. Toward this end, panels are simply flat surfaces of various areas from which heat rays may depart in vast multitudes and without obstruction. If the area of departure is extended sufficiently, every corner of a room may be bombarded effectively.

The practice of using lower ambient temperatures in a room that is panel heated generates from the ability of radiant energy to warm our receptive skins without regard for the temperature of the surrounding air. This fact, however, is strictly limited in application, since we cannot always be sure of receiving all the radiant heat we need as the result of chairs, tables, draperies, and even the bulk of other persons coming between us and the source of the confortable rays. In consequence of this vulnerability to temporary or spotty chilling where radiant panels are concerned, we should limit the drop in our ambient design temperature to 4 or 5 deg below the choice for other methods of heating. Thus a 74 F room for steam or hot water becomes a 69 F or 70 F room when served by radiant panels.

Physiological Factors. Operational reports from panel installations present much conflicting evidence. For example, garage mechanics often declare themselves pleased by warm floor panels which apparently reduce the incidence of rheumatism, whereas office workers and housewives just as often complain of swollen feet and ankles. Again, ceiling panels will be eminently satisfactory to some persons, while others are sensitive to an annoying impact of heat on their heads.

From all data currently received, we are justified in drawing the following conclusions:

1. Some persons are more sensitive to warm floors than others. To be safe, floor-panel temperatures should never exceed a normal maximum of 85 F, and 80 F is even more desirable.
2. Ceiling-panel temperatures, for ceilings about 8 ft high, should be kept lower than 110 F.
3. Temperature limits for wall panels, 115 F maximum, 110 F desirable.

Other Factors. Panel installations are sensitive to the placing of rugs, draperies, and furniture. Infrared rays from adjacent buildings and from the stratosphere, entering through the windows, have been known to upset a room's balance.

Furnishings should be considered somewhat in designing a system, particularly where floor coverings are concerned, but any influx or rays from exterior sources normally should be ignored. Such bonuses as solar heat are limited and unpredictable.

A serious consideration is ventilation. Because of the number of air changes required in schoolrooms and auditoriums, any panel system designed to meet more than two air changes per hour is likely to require an excessive mean radiant temperature. Supplementary heating of some sort is therefore generally in order.

For ceiling panels, we should give thought to the plaster. Some plasters offer better emissivity than others; some crack and crumble at unacceptably low temperatures. T. Napier Adlam in his book on radiant heating recommends a plaster mixed and applied in accordance with the ASA A42-1-46 Standard Specification for gypsum plastering. Material over the pipes should be at least $\frac{3}{8}$ in. thick.

The Nine Steps of Panel Design

1. Selecting the Room Air Temperature. The room air temperature we select, as with all heating systems, is the basis on which we compute the heat load of the room. We already have observed that panel installations are entitled to a reduction of 4 or 5 deg in the ambient temperature as compared with more traditional systems; consequently we shall select our room air temperature on this premise.

2. Locating and Approximately Sizing the Panel. We decide if the panel is to be placed in the floor, in the wall, or in the ceiling, and where in these areas it will best fit.

Sizing the panel is not at all difficult—we just make it as large as practicable.

The advantage of a large panel is observed in the lower temperatures required of a widespread surface as compared with a restricted one; in addition, more uniform distribution of the heat is obtained, exactly as a number of evenly spaced lights provide better over-all illumination than a single large globe.

3. Calculating the Heating Load. Using a room air temperature selected in accordance with step 1, we are now ready to calculate the heat losses from each room in the project. This is done precisely as with any other method of heating.

The saving in heat that is hoped for in many radiant-panel designs is actually based on the difference between room air temperatures, for example, 74 F for a hot water system, and 70 F with radiant panels. This saving shows up in the total calculated heat loss in Btu per hour.

4. Determining the Mean Inside Temperature of the Walls. With an outside temperature of 0 F and a room air temperature of 70 F, we know that the inside surface of the wall will present a temperature somewhere between the two.

The calculation is a simple one. If the coefficient of heat transfer is 0.35, as taken from the building tables in Chap. 2, then for the 70 deg of temperature difference, each square foot of wall will pass 70×0.35, or 24.5 Btu per hr. If the usual 1.65 film coefficient of conductance for inside walls is assumed, the temperature difference between the wall's inside surface and the room air becomes $24.5/1.65$ or 14.6 deg. This indicates that we may expect a wall temperature of $70 - 14.6$, or 55.4 F.

Applying the same procedure to the glass and floor areas, we discover that each square foot of glass will present a surface temperature of approximately 22 F, a value low enough incidentally to deposit frost on the pane. The floor temperature, if the floor is directly on the ground, reflects the low heat losses in that direction, which result from the stability of ground temperatures. In this case, the surface of the floor will be a comfortable 69 F.

So far we have assumed that the heating panel will be placed in the ceiling; and for that reason a calculation of the ceiling surface temperature does not belong in this step, but must wait for step 6. Of course, if the panel were going into the floor, or a wall, then the ceiling's mean surface temperature would be calculated here, along with the other unheated surfaces.

5. Determining the Mean Radiant Temperature. We all know that the transfer of heat from a warm surface to a cooler one depends on the temperature difference between them; consequently, to calculate the heat transferred from our warm panel to the cooler surfaces of walls, glass, and floors, we could (1) consider each surface separately or (2) we could bring them altogether in a single relationship by use of the mean radiant temperature. This latter procedure is certainly the easier and quicker of the two, and we arrive at a value for the mean radiant temperature as follows:

Assume that we have a room 15 ft 0 in. by 12 ft 0 in. by 8 ft 0 in. high containing several windows which add up to a total glass surface of 60 sq ft. We now are ready to tabulate all our data and produce a mean radiant temperature for this particular room (see Table 9.1).

TABLE 9.1. DETERMINATION OF MEAN RADIANT TEMPERATURE

Surface	Area, sq ft	Surface temperature, F*	Numerical product
Outside wall	432	55.4	23,932.8
Outside glass	60	22.0	1,320.0
Floor	180	69.0	12,420.0
Total	672	....	37,672.8
Mean radiant temperature	37,672.8/672, or 56.1 F, approximately		

* Surface temperatures in this illustration are from step 4.

This mean radiant temperature will be used as the common temperature of the cooler surfaces of the room, to each of which the ceiling panel will dispatch its invisible rays of heat.

6. Calculating the Surface Temperature of the Heating Panel. At this point we refer to step 3 to see how many Btu per hour we must deliver to this room in order to keep it warm. Suppose that in this case our heat-loss calculations with a 70 F air temperature in the room indicate that we need 14,000 Btu per hr. Some of these 14,000 Btu will come straight from the ceiling in the form of radiant heat; the remainder will wander off on the wings of convection. The ASHVE recommends that we divide the 14,000 as shown in Table 9.2.

Our ceiling panel will therefore emit 14,000 × 70 per cent, or 9,800 Btu per hr by radiation.

To calculate how warm a ceiling is needed to transfer 9,800 Btu per hr to other surfaces having a mean radiant temperature of 56.1 F, we shall exercise the Stefan-Boltzmann radiation equation, which states that the transfer of heat is proportional to the fourth powers of the absolute temperatures of the surfaces involved. This equation usually is written

$$q = 0.174e\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]$$

where q = Btu/hr/ft^2 of area (the ceiling area in this case is 180 sq ft)
e = surface emissivity factor (for a black body this is 1; for a plastered wall or ceiling, 0.9; for glass, 0.9)
T_1 = absolute temperature of the warmer surface
T_2 = absolute temperature of the cooler surface (in this case, 56.1 + 460.0, or 516.1 F)

Since we know the quantity of heat emitted and the area of emitting surface, we can insert these factors in the equation and get

$$\begin{aligned}\frac{9{,}800}{180} &= 0.174 \times 0.9\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{516.1}{100}\right)^4\right]\\ 54.5 &= 0.156\left[\left(\frac{T_1}{100}\right)^4 - 709.47\right]\\ 349.4 + 709.47 &= \left(\frac{T_1}{100}\right)^4\\ 570\text{ F} &= T_1\end{aligned}$$

And 570 − 460 = 110 F is the ceiling surface temperature.

This surface temperature is about as high as we dare design for, in a room with a ceiling only 8 ft high. The surface temperature of the human forehead is about 93 F, and a surface 17 deg warmer and only 24 to 30 in. distant would create a distinct feeling of heat impact on a person in a standing position.

TABLE 9.2. ASHVE RECOMMENDATION FOR DIVISION OF REQUIRED 14,000 BTU

Location of panel	Per cent by radiation	Per cent by convection
Ceiling	70	30
Floor	55	45
Wall	65	35

If our load for this particular room were even slightly larger, we should be obliged to supplement the ceiling panel with some other source of heat, probably a wall panel or convector radiator.

Right here, we might examine the manner in which this same equation is used to determine the amount of heat that may be emitted, without too much complaint, from a radiant panel located in the floor. The maximum surface temperature acceptable is 85 F; consequently q is the unknown factor in the floor problem, and we set up the equation this way, assuming that the mean radiant temperature is unchanged,

$$\begin{aligned} q &= 0.174 \times 0.9\left[\left(\frac{545}{100}\right)^4 - \left(\frac{516.1}{100}\right)^4\right] \\ q &= 0.156(882.16 - 709.47) \\ q &= 26.94 \text{ Btu/hr/ft}^2 \end{aligned}$$

For our floor of 180 sq ft, this means a total emission of 4850 Btu, approximately, from radiation. Since radiation from a floor panel represents 55 per cent of its total heating value, 45 per cent being chargeable to convection, this particular floor will deliver in all ways 4850/0.55, or approximately 8820 Btu per hr. Obviously, this is about 1000 Btu per hr less than we need; therefore we must provide some auxiliary means of heating in the circumstances.

While wall panels permit the use of surface temperatures considerably higher than the temperatures that are acceptable for floors, and sometimes for ceilings also, the panel area for a wall installation is almost always limited by the presence of doorways, closets, draperies, and windows. For this reason, wall panels that seem quite suitable at first thought often turn out in the calculations to fall short of our needs. The procedure for calculating wall-panel surface temperatures is exactly the same as that for ceilings and floors. Only the proportioning of the radiant and convection emissions is different, as we have already noted in this step, being 65 per cent for radiant energy, 35 per cent for convection.

7. Selecting the Heating Agent. Although this step appears here as the seventh in the design sequence, most of us will know before we start on a job whether we intend to use hot water in pipe coils, warm air in ducts, a hollow ceiling, or electricity in embedded cables. Each of these agents presents its own singular problems in the construction of the building, and these requirements and problems normally are settled before the actual heating design commences, usually through conferences with the architect or owner.

However, each of the six steps that proceeds this particular one must be taken irrespective of the heating agent, and we can if we wish, or find it expedient, postpone the selection of the agent until this time. But from this point, the design road branches, and we shall examine each of these branches in turn, beginning with step 8.

8. Sizing the Water Coils. When we undertake to heat our panel with circulating hot water, these four factors must be considered as affecting the arrangement of the piping:

a. The material in which the piping is embedded naturally affects the diffusion of heat from the pipe surface to the panel surface.

b. Depth of cover, between the pipe surface and the panel surface, affects the speed of heat transmission.

c. Spacing of the pipes with relation of one to another affects the surface temperature and its uniformity.

d. There is a loss of heat *away* from the panel. Thus ceiling pipes emit heat upward as well as downward, floor pipes heat the ground or basement to some extent, and wall pipes dispatch energy outward as well as inward.

Suppose that we consider these factors in order. Ceiling pipes generally are buried in plaster, the plaster having a total thickness of $\frac{3}{4}$ in. Since the conductivity of plaster is low, a thin layer over the pipes exercises a notable insulating effect; consequently, water temperatures up to 140 F usually are safe, and the ceiling surface will not be cracked from overheating. Some designers even circulate water of 150 F through ceiling coils without experiencing any trouble.

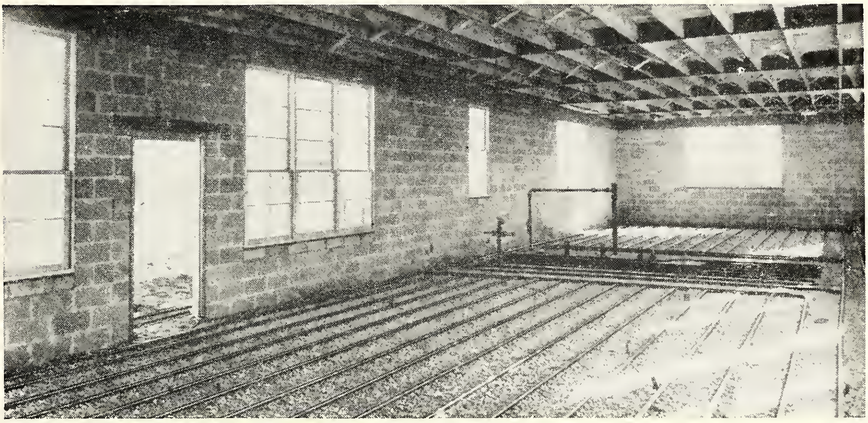


Fig. 9.1. Radiant heating piping in form of floor grid. (*Courtesy of A. M. Byers Company.*)

Down in the floor, the conductivity of the concrete is comparatively high, and a layer of concrete 2 in. thick over the pipes is therefore common practice. The construction procedure generally places from 6 to 8 in. of broken stone on the ground, then the pipe coils are laid on the broken stone, and the concrete slab is poured either with or without additional stone being dropped between the pipes. Sometimes a cork layer is laid under the pipes or over the pipes, depending on whether we are trying to arrest the escape of heat upward or downward. These cork layers will approximate 2 in. in thickness, and should be separated from actual contact with the hot pipes by an air space. Alternately, rock wool, asbestos, or types of insulation may be employed. Figure 9.1 illustrates a common floor coil installation.

How we space the pipes depends largely on the amount of heat we must provide. According to research of the ASHVE, we can expect the heat emissions listed in Table 9.3, expressed in Btu per foot of pipe per degree of temperature difference between the water and the room air.

TABLE 9.3. HEAT EMISSIONS

Pipe size, in.	Center to center spacing, in.*	Emission, Btu/ft/hr/deg temp. diff.	
		Concrete	Plaster
½	6	0.80	0.72
¾	9	1.00	0.90
1	12	1.20	1.08

* *Note:* By widening the spacing between pipes, we can increase the rate of heat emission up to approximately 10 per cent, until the distances are about double those given. Beyond that distance, no increase is observed.

Now we may estimate the length of tubing required for our illustrative job, which demands an emission from the coil of 14,000 Btu per hr. We shall assume that several inches of rock wool is dumped on top of the coil to prevent heat escaping in the upward direction, and with this sort of construction 90 per cent of the heat will emerge from the panel, as we desire. However, in order to harvest a full 14,000 Btu each hour from the panel, we must provide compensation for the upward loss of 10 per cent; consequently the actual input to the coil will be 14,000/0.9, or 15,556 Btu per hr.

For the ceiling panel, we are almost obliged to use ½-in. pipe in order to keep within the thickness of the plaster; and since the effective emission of a pipe of this size on 6-in. centers is 0.72 Btu/ft/hr/F of temperature difference, we shall obtain (140 F − 70 F) × 0.72, or 50.4 Btu per hr from each foot of the pipe. With a total input demand of 15,556 Btu per hr, we shall need 15,556/50.4, or 309 lin ft. These 309 ft, if installed in the form of continuous tubing, will just about cover the ceiling area, with 6 in. between the center lines of the parallel pipes.

An annoyance that sometimes follows the installation of a ceiling coil is the development of dirt streaks on the ceiling. These streaks appear on the cooler areas between the tubes, and are the result of a thermal deposition of dust, comparable to the streaks that appear on walls behind conventional steam radiators.

To help avoid these undesirable streaks, we should endeavor to maintain as uniform a surface temperature as we can, and one step in this direction is to arrange the coil somewhat in the form of a reversed return system. Figure 9.2 shows a ceiling panel arranged in this manner. Figure 9.3 illustrates a typical wall installation.

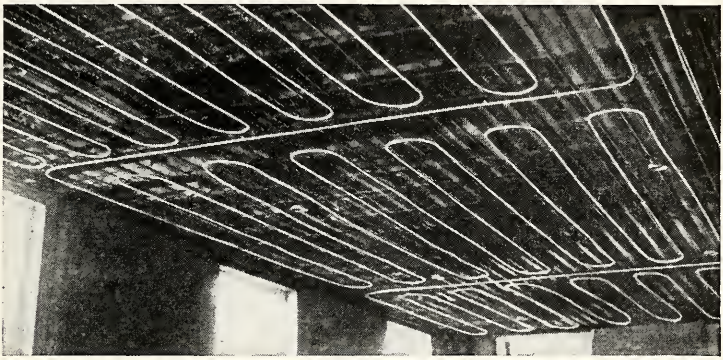


Fig. 9.2. Radiant heating copper tubing in ceiling. (*Courtesy of Chase Brass & Copper Company.*)

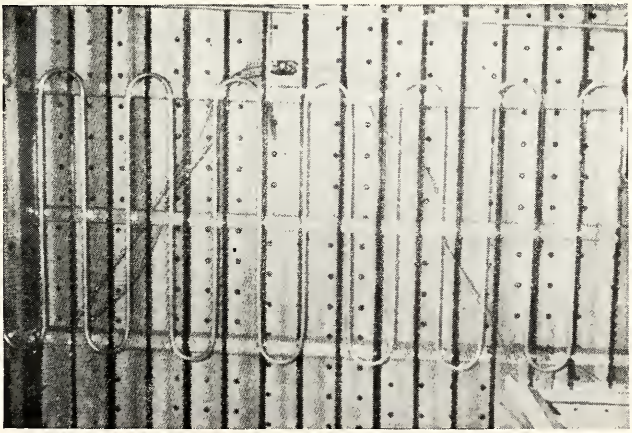


Fig. 9.3. Radiant heating copper-tubing wall. (*Courtesy of Chase Brass & Copper Company.*)

9. Venting and Controlling the System. Unless we are using air as the heating medium, we want none of it in a heating circuit. At this point in the design, therefore, we scan our piping layout for high spots at which to place automatic air relief valves. This venting should take place either before or after the coil, never in the coil itself, since a water leak in a ceiling can be a very messy affair and drip hot discredit on the designer.

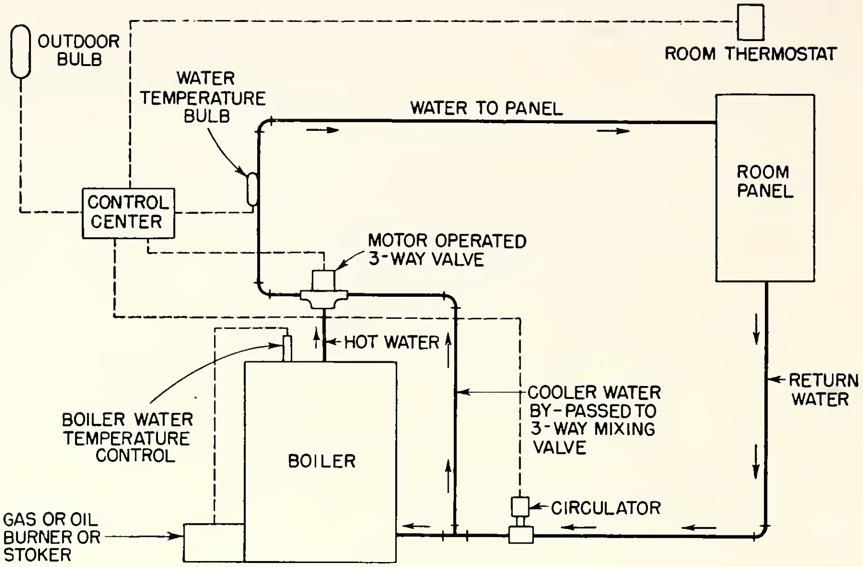

OUTDOOR BULB
ROOM THERMOSTAT
WATER TEMPERATURE BULB
WATER TO PANEL
ROOM PANEL
CONTROL CENTER
MOTOR OPERATED 3-WAY VALVE
HOT WATER
RETURN WATER
BOILER WATER TEMPERATURE CONTROL
COOLER WATER BY-PASSED TO 3-WAY MIXING VALVE
BOILER
CIRCULATOR
GAS OR OIL BURNER OR STOKER

Fig. 9.4. Arrangement of controls to vary water temperature being delivered to radiant coils.

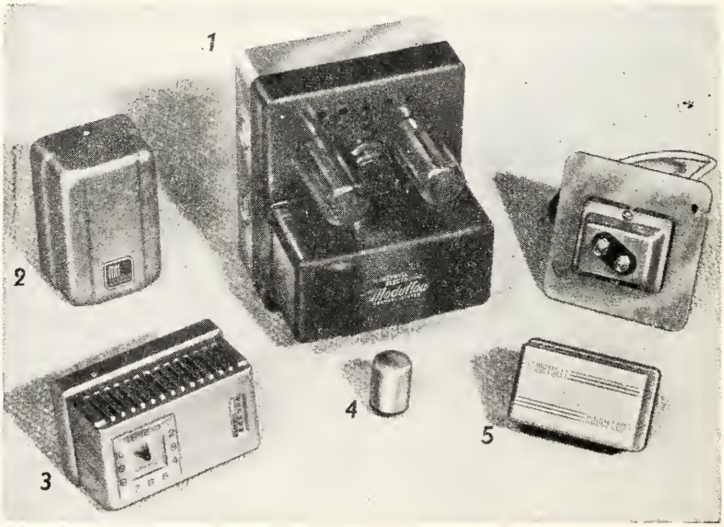


Fig. 9.5. Electronic controls for use with water or warm air ceiling panels. 1. Relay amplifier. 2. Outdoor anticipator. 3. Electronic Chronotherm. 4. Cycler. 5. Averaging thermostat. (*Courtesy of Minneapolis-Honeywell Regulator Co.*)

Controlling a radiant panel involves our consideration of the element of heat lag. For example, 4 in. of concrete may require up to 24 hr for thorough cooling, or thorough heating. In consequence of this situation, the best plan for heat control will incorporate some form of outdoor weather anticipator which works in cooperation with an indoor thermostat.

Figure 9.4 illustrates such an arrangement. The moving center of the control is a three-way valve that mixes hot water from the boiler with the cooler return water before delivery is made of the blended water to the panel coils. The proportions of hot water and return water are varied by the thermostatic bulb of the outdoor instrument, with the indoor thermostat functioning as an overriding monitor to prevent either overheating or underheating.

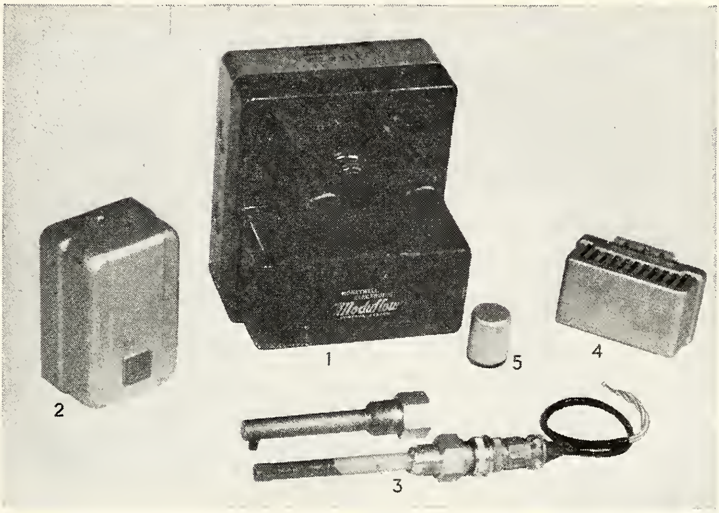


Fig. 9.6. Electronic controls for service with water or warm floor panels. 1. Electronic relay amplifier. 2. Outdoor anticipator. 3. Immersion thermostat. 4. Electronic thermostat, mounted in room. 5. Cycler. (*Courtesy of Minneapolis-Honeywell Regulator Company.*)

As an illustration of this scheme of control, suppose that we are in a comfortable room when a sudden drop of the outdoor temperature causes the outdoor instrument to increase the admission of hot boiler water to the system. Almost certainly, save for the indoor monitor, we should be overheated, since the thermal losses from the room lag well behind the sensitivity of the outdoor thermostat; and this condition of overheating would continue for some time, perhaps hours, until room losses and coil input came into balance.

Overheating does not develop, however, because our indoor monitor either prevents or restricts the admission of extra hot water to the coil, until the room temperature itself shows some change as the result of the weather's mutation.

The indoor thermostat for radiant panel service is sometimes called a "comfortstat," and differs from the common instrument by being sensitive to radiant rays. Figure 9.5 shows one of these comfortstat-type combinations for ceiling panels, and Fig. 9.6 for floor panels. Figure 9.7 provides a close-up of the inside instrument.

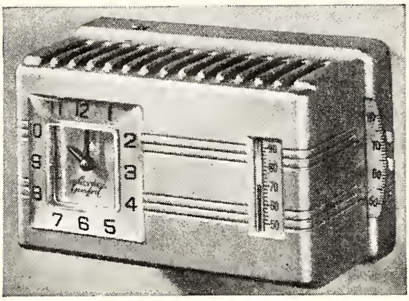


Fig. 9.7. Close-up of Chronotherm for ceiling panels. (*Courtesy of Minneapolis-Honeywell Regulator Co.*)

Warm Air Panels

Characteristics. The earliest form of panel heating on record seems to have employed a warm air arrangement by which hot flue gases were directed through floor and wall ducts in the ancient Roman baths. Nowadays we use clean warm air instead of the products of combustion, and the circulation takes place behind a ceiling, the fan-propelled air moving from side to side of the room between sheet-metal guides.

One great advantage of this modern warm air panel is seen in its freedom from streaking; one disadvantage appears in its construction requirement for approximately 4 in. of clear air space between the radiant panel and the closing-in surfaces overhead. To provide these 4 in., we must choose between a lowered ceiling and a raised roof, the former being undesirable usually, and the latter adding to the over-all cost of the structure.

Procedures of Design. The first seven steps previously described must be taken in the design of every panel system. The eighth step for warm air, proceeds as follows:

1. Locate both the supply and return ducts.

2. Size all the ducts, observing the rules explained in Chap. 8 which govern mechanical warm air heating. As with other forms of warm air, building characteristics may oblige us to use wide flat ducts in single or multiple, in order to deliver the warmed air to the ceiling. Large rooms almost invariably will need two or more ducts in the supply system.

3. Figure 9.8 shows a typical warm air ceiling with the air guides in place. These guides should be of 30-gauge metal, or heavier, and spaced at least 1½ times the width of the duct at the point where the duct enters the panel.

4. If possible, arrange the duct connections and guide strips to direct

the entering air first along outside walls, thereby meeting the area of greatest cold with our hottest air.

5. Keep all ducts as short as possible, and attempt to bring together in a common plenum the returns from all the panels. A design of this sort simplifies the installation and conserves cost.

6. Insulate all ducts in outside walls.

Control of Warm Air Panels. When the fan that circulates the warm air is stopped, the heating stops also; there is no reservoir of Btu still present as in the case of a circulating water system.

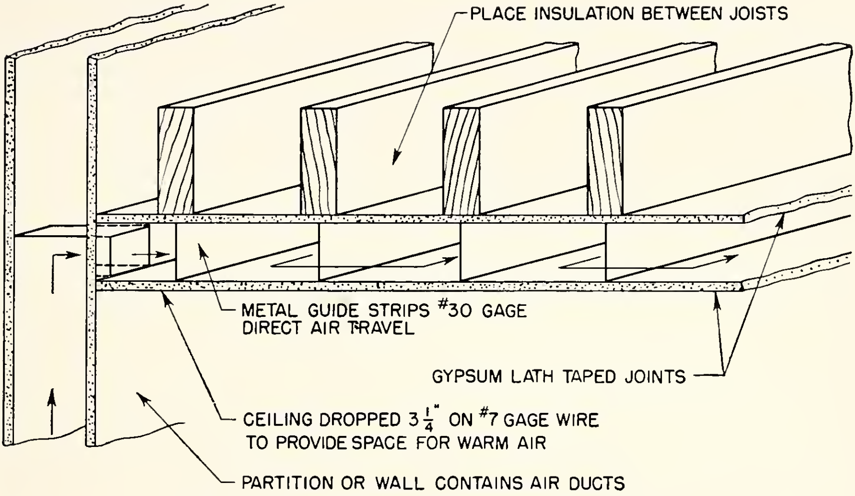

PLACE INSULATION BETWEEN JOISTS
METAL GUIDE STRIPS #30 GAGE
DIRECT AIR TRAVEL
GYPSUM LATH TAPED JOINTS
CEILING DROPPED 3 1/4" ON #7 GAGE WIRE
TO PROVIDE SPACE FOR WARM AIR
PARTITION OR WALL CONTAINS AIR DUCTS

Fig. 9.8. Arrangement of warm air ceiling panel.

Because of this characteristic, a modulating form of control which varies the temperature of the air being served to the panel is preferable to a flow-interrupting arrangement. The latter scheme obviously tends to make ceiling temperatures swing across wide extremes as the only alternative to a continuous off-and-on operation of the fan and firing equipment which definitely increases maintenance.

The controls described as being applicable to hot water panel systems may be adapted to warm air designs, simply by actuating a duct damper rather than a three-way water valve. By means of this damper, we readily recirculate some of the return air, thereby reducing the air temperature to the panel, during periods of low heat demand.

Electric Panels

Characteristics. Since the use of electric cables in the floor or ceiling to create radiant panels is seldom popular except in areas of low-cost current, we find most of these installations in the Pacific Northwest and in the Tennessee Valley.

The conductor cable is small, about 1/8 in., and is designed for both 110- and 220-volt circuits. The maximum operating temperature of the conductor varies with the manufacturers of the panel systems, usually from 135 to 167 F.

Cables are installed in the ceiling or in the floor, just as water piping is installed, but the electric arrangement is always in the form of a continuous coil, whereas some of the water designs take the form of a grid.

Sizing Electric Panels. After calculating our room heat losses in the usual way, we can determine the length of cable required by means of Table 9.4. The cables may be spaced on 4-in. centers, or greater, depending on the heat concentration permissible in the ceiling. As with water and air panels, we do not desire to crack or spall the plaster. We can calculate the surface temperature in the usual way, employing the Stefan-Boltzmann radiation equation, with the T_1 temperature as the unknown factor.

TABLE 9.4. BASES FOR DETERMINATION OF LENGTH OF CABLE REQUIRED

Voltage	Standard length of cable, ft	Output, watts
110	60	250
110	500	1,500
220	120	500
220	1,000	3,000

For some estimate of the operating costs, useful in a general way prior to the actual calculation, we may assume that small rooms will consume 2 watts per hr per cu ft of room volume, large rooms 1½ watts per hr.

Preformed Panels. It is possible to buy ready-formed panels in which the heating element is sandwiched between layers of wallboard. The standard design consumes 25 watts for each square foot of panel area, but greater capacities may be obtained on special order. Surface temperatures of the panel normally approximate 120 F, with the inside element operating at 135 F.

Preformed panels present a problem of expansion since they spread when heated. In one solution, we simply cover all joints with battens, but this expedient requires us to sacrifice a smooth ceiling. Under an alternate plan, we install the panels around the perimeter of the room. Then the panels are heated for 48 hr to be sure that the maximum expansion is reached, and allowed to cool, after which the inside edge is nailed in position against the unheated ceiling. The open joint is then sanded, filled, and taped, exactly as gypsum-board joints are smoothed and sealed.

Another form of panel is built with electrically conductive rubber sealed in between layers of insulating fabric. Stiffening is provided by a backing of asbestos board. These panels are interesting in that no wires are employed in the panel, the conductive rubber performing the dual duty of current conductor and heat-emitting element.

Standard ratings are approximately 22 watts per sq ft.

Radiant Glass Panels. In these units, a heating element of aluminum is fused between sheets of tempered glass, and is backed up with a polished aluminum reflector. Glass panels are quite small, being rated generally from 500 to 1,000 watts each.

In the moderate climate of the South, some homes are being heated entirely with glass panels, one or two being installed under strategic windows in each room, as necessary.

The current consumption of these units may be estimated for an entire heating season from the following equation:

$$\text{Total kilowatthours} = \frac{\text{total cubic feet of volume} \times \text{degree-days} \times 0.2}{1{,}000}$$

Baseboard Heating

Characteristics. In some designs, the baseboard fits close to the wall, and there is a more or less limited movement of air to provide convection heating. In other designs, the baseboard incorporates a finned coil, projects outward from 3 to 4 in. from the wall, and emits up to 1200 Btu per hr per lin ft.

This latter design is seen mostly in public places rather than residences, since the substantial projection defeats the inconspicuousness most desired in a home.

Residential baseboard becomes a problem at times, because it is hard to make the comparatively limited heat emission meet the restrictions imposed on its length by doorways, French windows, fireplaces, and so on. Meeting the entire load with baseboards only is sometimes impossible.

This sort of heating apparatus is not cheap to install, but it does respond rather quickly to weather changes. When operating with water, the temperature should be kept below 200 F, to inhibit the formation of dust streaks on the wall above the baseboard. With steam, we must accept the streaks.

A cast-iron residential baseboard is illustrated in Fig. 9.9; the finned-tube type appears in Fig. 9.10.

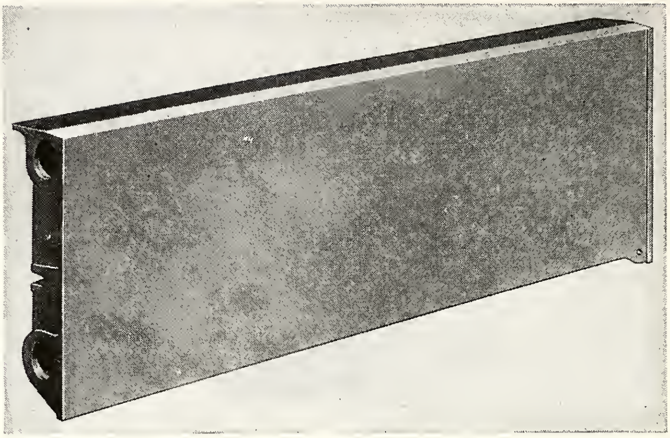


Fig. 9.9. Cast-iron baseboard for residential installation. (*Courtesy of American Radiator and Standard Sanitary Corporation.*)

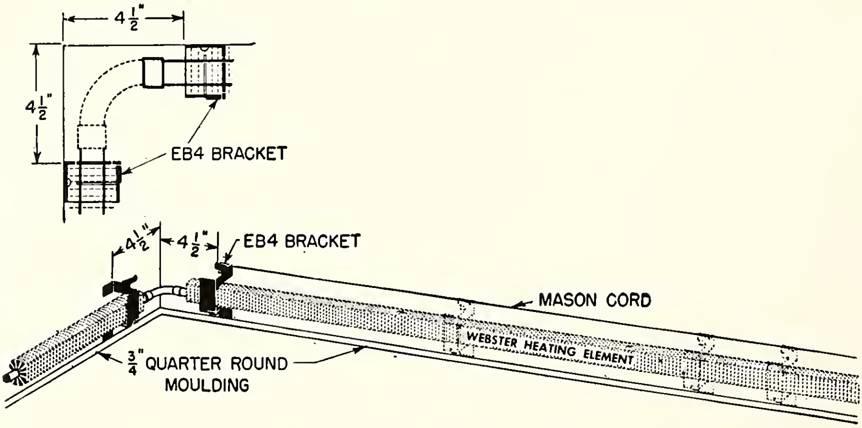


Fig. 9.10. Finned-type of baseboard. Note that air movement over finned tubing is thrown outward to avoid wall discoloration. (*Courtesy of Warren Webster Company.*)

Radiant Cooling

There are radiant heating installations here and there throughout the country, to which cold water is delivered in summer to provide cooling.

By circulating water at 45 F, panel temperatures of 52 to 55 F are obtained, and these panels will absorb from 1.2 to 1.5 Btu/hr/ft^2/F of temperature difference between the panel and the room air.

As with heating, the panel surface temperatures are limited, not only by the restrictions of human tolerance but also by the dew point of the room atmosphere as well. Floor temperatures of 70 F appear to be the lowest recommended by experienced designers, cooler floors tending to aggravate rheumatic conditions. Ceilings are free of this restriction, but must be controlled with care in order to remain safely above the dew point.

Because of the narrow spread between panel surface and ambient air temperatures, radiant cooling installations may require auxiliary apparatus to provide satisfactory comfort for periods of peak demand. They also require much of the time, except perhaps in the arid regions of the country, an air dehumidifying system of some sort, in order to avoid panel surface condensation.

In view of the considerable supplementary machinery so often necessary for radiant cooling, the economic balance tends to become unfavorable. The added cooling benefits, if any, from a radiant heating design probably will vary widely with individual installations; consequently each job must be weighed rather critically before summer cooling is recommended by the heating designer.

Unit Heaters

Characteristics. A unit heater is simply a finned pipe coil through which hot water or steam is circulated, a fan being located behind the coil to deliver warm air at an accelerated rate to the space being heated. One alternate form should be noted, namely, the gas-fired unit, in which the pipe coil is replaced by flues through which the products of combustion pass off to the outside air. The air from the fan is heated by its passage across these flues. A conventional form of the unit heater is illustrated by Fig. 9.11; the gas-fired form by Fig. 9.12.

Applications and Installations. The unit heater is especially suitable for heating large spaces such as gymnasiums, stores, auditoriums, and restaurants. Not being a creation of beauty, we sometimes endeavor to conceal its homely and occasionally noisy presence behind partitions and walls, the warm air then emerging to the room through a decorative grille. This plan is effective for churches and sometimes for schools.

For auditoriums and factories with high ceilings, the unit heater comes into its own, since it directs heat where heat is needed, and is in no way concerned with the height of the ceiling. Its own mounting generally is most efficient at a height of from 11 to 15 ft above the floor.

The arrangement of unit heaters is best solved by a study of the floor plan of the space to be heated. Since our blast of air is almost as direct and straight as the infrared rays of a radiant heating panel, we must make sure that the delivery of heat is not blocked by walls or other solid structures. We do not like to impose hot blasts on working personnel either, which means that a fair diffusion of the warm air must be achieved before the stream makes impact on persons in the line of flow.

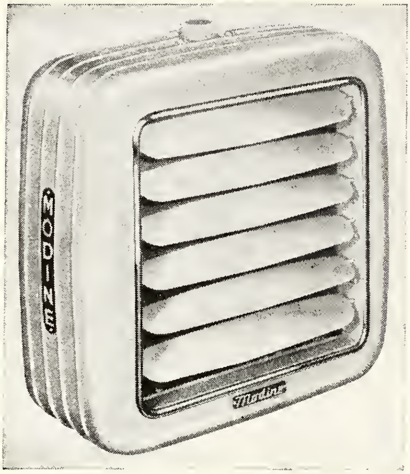


Fig. 9.11. Unit heater suitable for either steam or hot water service. (*Courtesy of Modine Manufacturing Company.*)

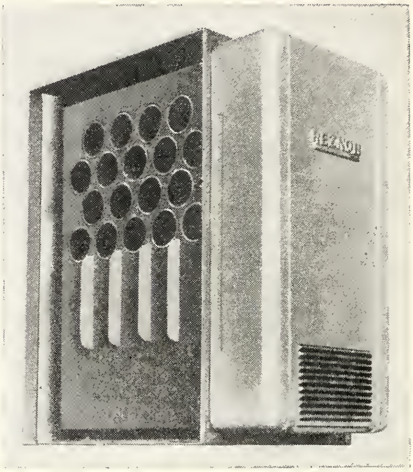


Fig. 9.12. Special gas-fired unit heater for installation in duct. (*Courtesy of Reznor Manufacturing Company.*)

A sound plan calls for the heaters to be so arranged that the general movement of the air in any given space tends to sweep the peripheral walls. In this way, heat is delivered to the outside surfaces that most need it, and the comparatively still areas within the moving border streams are protected from chill by the warm barriers thus provided.

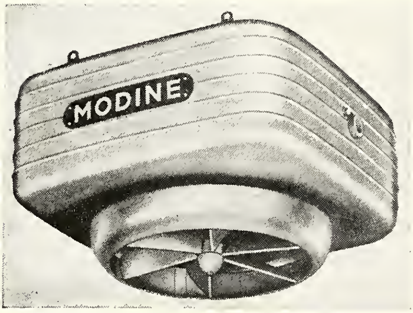


Fig. 9.13. Downblast type of unit heater. (*Courtesy of Modine Manufacturing Company.*)

The downblast type of unit heater, shown in Fig. 9.13, frequently is used to curtain with warm air those outside doors subjected to continuous opening and to being left open. This application commonly is seen in warehouses and factories where considerable traffic occurs between heated areas and loading platforms.

Figure 9.14 illustrates another style of unit heater; this type employs revolving discharge hoods for distributing the warmed air. In effect, the spaces are sprayed with heat.

In size and weight, unit heaters range from about 12 in. square to 60 in.

square, and from 30 lb to more than 600 lb in weight. In capacity, we may expect to find available standard units ranging from 8000 Btu per hr to more than 500,000 Btu per hr. The capacity varies of course with the temperature of the heating medium, and with the temperature rise imparted to the air as it makes its passage across the coils.

Control of Unit Heaters. A simple room thermostat will perform satisfactory service in most installations. This thermostat may control a single heater, or a battery of heaters, as desired.

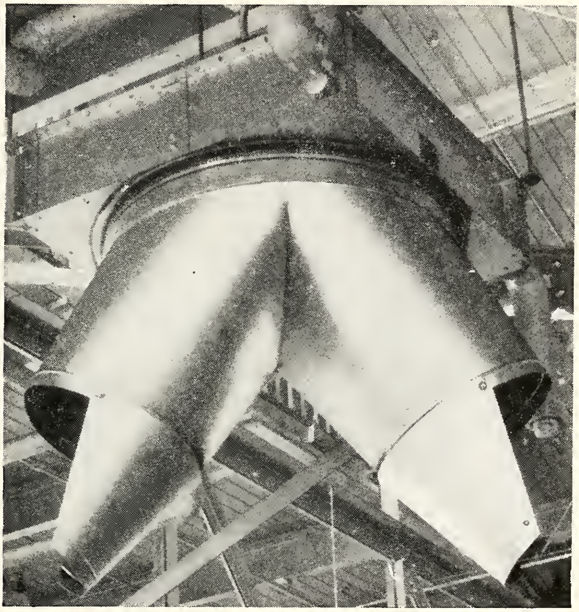


Fig. 9.14. Revolving type of unit heater, using steam or hot water. (*Courtesy of L. J. Wing Manufacturing* Co.)

In choosing gas-fired units, we should be sure they bear the American Gas Association stamp of approval before acceptance.

Questions

1. Describe the principle of radiant panel heating.
2. What physiological factors must we consider in designing a system?
3. Enumerate some of the material factors that affect the design.
4. What are the nine steps of panel design?
5. What room temperature would you select for a radiant installation in your own city?
6. Consider the installation of radiant panels in your own home. Would you use ceiling, floor, or wall panels? A combination? Why?

7. Describe the procedure by which we determine the mean inside temperature of any given room. What is the mean temperature in your home?
8. What is the mean radiant temperature? Why is it important?
9. How do we arrive at the surface temperature of the radiant panel?
10. Explain each factor in the Stefan-Boltzmann equation.
11. Why do floor coils alone frequently fall short of meeting the heating demands for a room?
12. If you were obliged to use floor coils for your basic heat-emitting unit, how would you supplement them to meet the total load?
13. For practice, assume that you will install water coils in your own home, and size them, using the procedure described in step 8.
14. How would you control a radiant panel system? Why?
15. What are the advantages of a warm air panel, if any?
16. Enumerate the six steps in the design of warm air panels.
17. How would you control a warm air panel system?
18. What are the advantages, if any, of the electric panels?
19. Of the electric panels discussed, which appeals most to you? Why?
20. Where and why would you use unit heaters?

FANS AND FILTERS

Fans

The National Association of Fan Manufacturers. The NAFM, which includes in its membership most of the important builders of fans, has set up standards for the physical proportions of fans, and, in addition, has classified the air-moving equipment.

This serious attempt to bring uniformity to the building of fans is of distinct value to the designer and to the user, for it makes possible a common assurance of minimum quality and performance.

Common Fan Characteristics. Certain basic laws govern the performance of fans, of which persons concerned with heating should understand at least the following:

1. The capacity of a fan varies directly with the speed. Double the speed, and the output in cubic feet per minute is also doubled. Halve the speed, and we halve the output.
2. Delivery pressure from a fan varies as the square of the speed.
3. Fan horsepower varies as the cube of the fan speed.

A knowledge of these laws is useful. For example, we may be tempted from time to time to increase the speed of a fan to meet demands somewhat beyond the standard rating. However, when we observe what this speeding up does to the fan horsepower, we may be forced to abandon the idea.

Calculating Fan Horsepower. Usually, we will refer to a fan manufacturer's catalogue, and note therein the horsepower required to solve our problems. However, when preliminary studies are in progress, a knowledge of horsepower calculations becomes useful. The air-horsepower equation is

$$\text{Air horsepower} = \frac{\text{cfm} \times \text{head of air, ft} \times \text{density}}{33{,}000}$$

$$\text{Head of air, ft} = \frac{\text{pressure, in. W.G.} \times 62.4}{12 \times 0.07488}$$

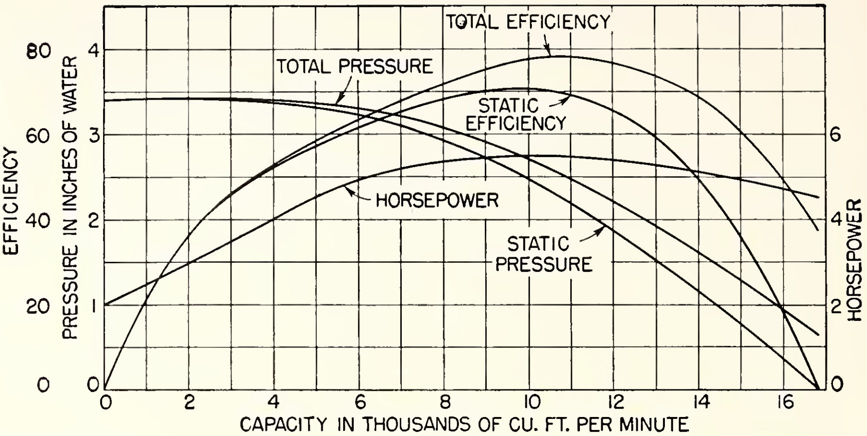


Fig. 10.1. Characteristic performance curve of backward curved fan. (Courtesy of *Buffalo Forge Company.*)

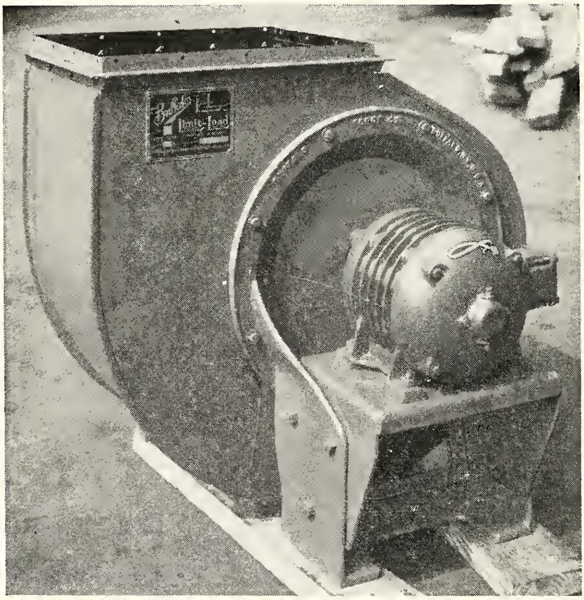


Fig. 10.2. Centrifugal type of blower, direct-connected to motor. (*Courtesy of Buffalo Forge Company.*)

These formulas may be combined into a single short one:

$$\text{Air horsepower} = 0.0001575 \times \text{cfm} \times P$$
$$P = \text{pressure, in. W.G.}$$

Air horsepower of course assumes a fan efficiency of 100 per cent. Each fan has its own characteristics, which may be set down in graphic form, as illustrated by Fig. 10.1. On these curves, we observe that even 80 per cent efficiency, when approached at all, may be expected across a very narrow range of capacity.

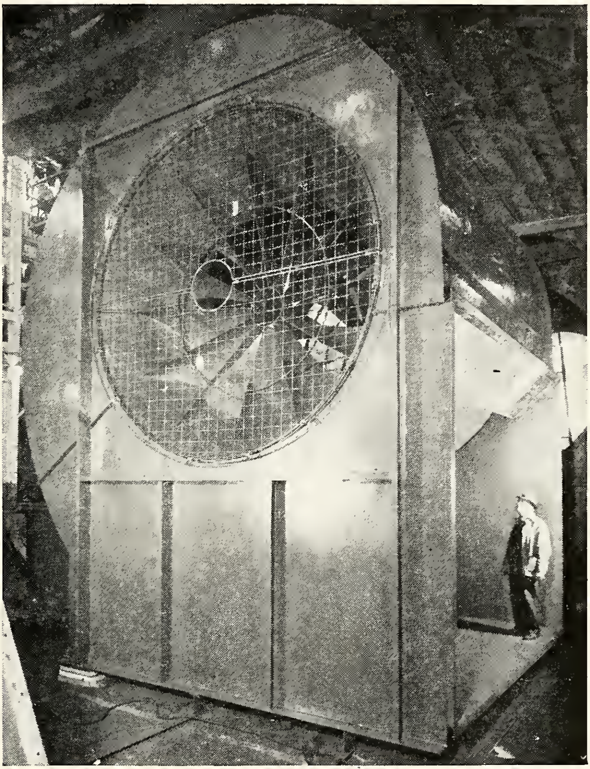


Fig. 10.2*a*. Large fan (*Courtesy of Buffalo Forge Company.*)

In the absence of any guiding curves, we must use our own judgment for efficiency, perhaps 50 per cent being acceptable for estimating the size of the driving motor.

Types of Fans. The NAFM classification pictures four basic fans in which heating engineers are likely to find their greatest interest.

Figures 10.2 and 10.2*a* show centrifugal fans, or blowers, which we use for large ventilating and air-distribution systems. This design also serves for forced and induced draft systems on boilers, and for the pneumatic conveying of various industrial materials, dusts, etc.

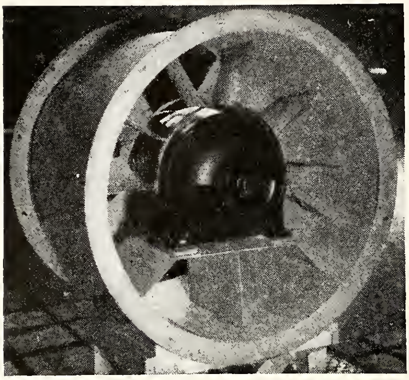


Fig. 10.3. Vaneaxial fan. (Courtesy of *Buffalo Forge Company.*)

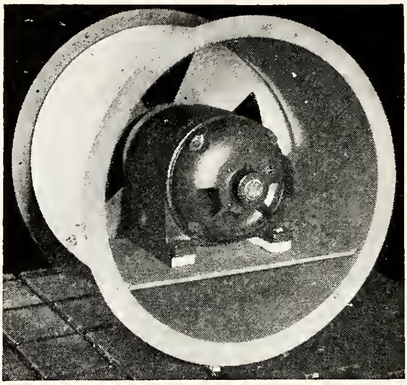


Fig. 10.4. Tubeaxial fan. (Courtesy of *Buffalo Forge Company.*)

Figure 10.3 presents the vaneaxial fan, a design that sets the motor in the duct. This is not a bad idea in certain cooling problems for the air bath assuages motor temperatures. These fans are used for ventilation in close quarters, such as aboard ship, in mines, and in many industrial processes. Vaneaxial fans may be slightly more noisy than centrifugal units of comparable capacity.

Figure 10.4 illustrates the tubeaxial fan. It is similar in design and characteristics to the vaneaxial design, but is built without the stationary directional vanes that are observed in the latter fan.

Figures 10.5 and 10.5*a* indicate the basic appearance of the propeller fan, which commonly is used for ventilation without ducts. We find propeller fans ventilating home kitchens, attics, commercial establishments, and industrial compartments. This fan is not suitable for duct installation, since its design characteristics do not provide a high-pressure discharge. It is suitable for moving large volumes of air against low resistances.

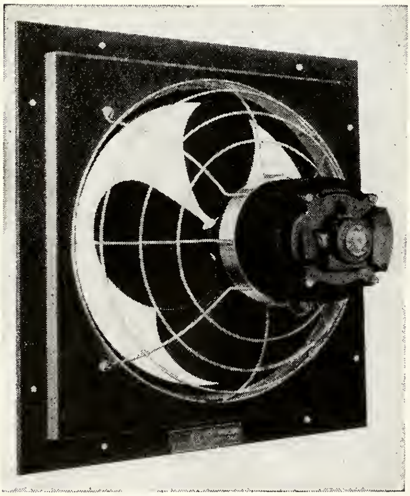


Fig. 10.5. Propeller fan, belt-connected. (Courtesy of *New York Blower Company.*)

Selecting a Fan Size. The catalogues of the fan manufacturers tabulate all the pertinent data that we need for making a selection. As an illustration of how these data are presented, a representative group is shown in Table 10.1.

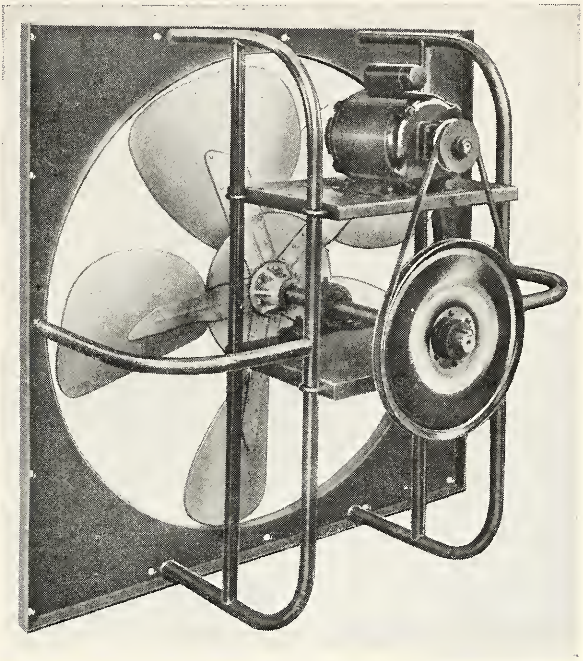


Fig. 10.5*a*. Propeller fan direct-connected. (*Courtesy of New York Blower Company.*)

TABLE 10.1*

Outlet velocity, fpm	Capacity, cfm	¼-in. S.P.		½-in. S.P.		1-in. S.P.	
		Rpm	Hp	Rpm	Hp	Rpm	Hp
1,300	21,734	316	1.81	366	2.72	450	4.75
1,900	31,765	420	4.33	455	5.48	525	8.09

* *Note:* Special conditions are noted at the bottom of each manufacturer's table. For example, in this group, the horsepower refers to actual fan horsepower only. When selecting the motor, we must add for efficiency loss through the drive and in the motor. Capacities generally are based on air at 70 F and a 29.92-in. barometer. If our air conditions and elevation vary far from these standards, we must correct the table accordingly.

Fans for Corrosive Service. Oftentimes, and particularly in chemical industries, the heating engineer finds himself faced with the need to move fume-laden air. Before selecting a fan in this situation, we must determine if the fumes are corrosive, and specify a fan that is resistant to the particular corrosion present.

The fan manufacturers are prepared to supply fans with cast metal housings and with protective coatings of lead, rubber, enamel, or whatever is needed on both housing and wheel.

Filters

Discussion of Types. The type of filter most common in heating design and practice is the dry-mat form, illustrated in Fig. 10.6. In this unit, the frame contains porous or fibrous material through which the air is forced, and since the openings are too minute for many of the air-borne motes to pass, the air emerges well cleaned of the undesirable matter.

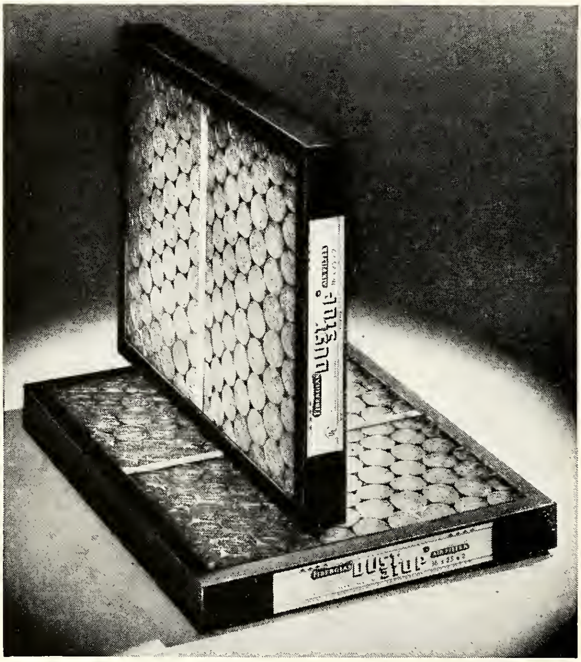


Fig. 10.6. Dry-mat type of filter. (*Courtesy Owens-Corning Fiberglas Corporation.*)

Obviously these labyrinthic capillaries clog in time under an accumulation of debris. When this occurs, we throw away the filter (hence the commercial term "throwaway filter") and replace it with a new one. Cleaning of dry mats seldom is either practical or desirable economically.

On warm air heating systems, this type of filter is practically a standard. Replacements at the beginning of each heating season usually meet all requirements for filtered air, but in some industrial or commercial locations the presence of air-borne lint will stop up a dry-mat filter in a few weeks. Incidentally, the handling of lint is one of the most difficult of all filtering problems; the stuff not only closes air passages in a short time but it also resists removal with extraordinary success.

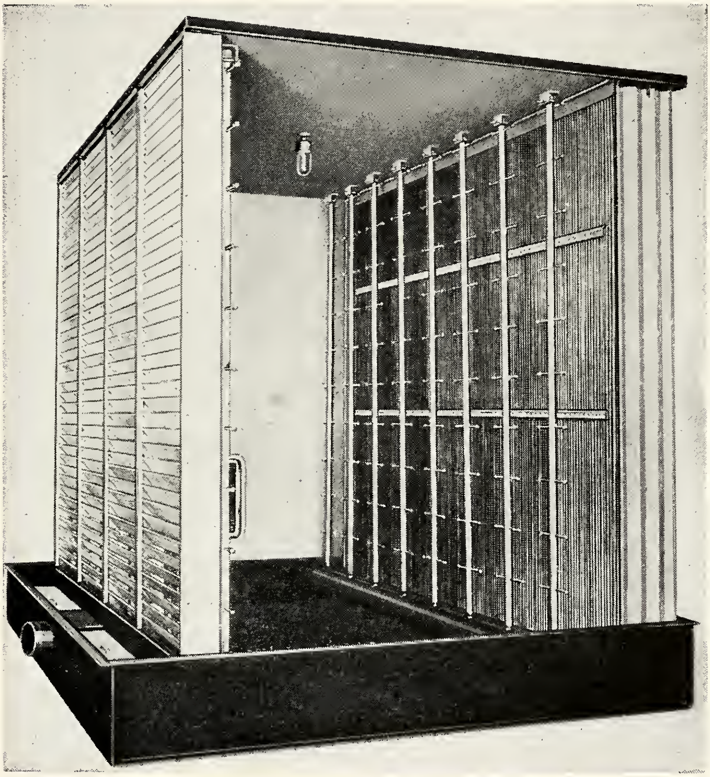


Fig. 10.7 Spray washer. (*Courtesy of American Blower Corporation.*)

The *viscous filter* is shown in Chap. 6, Fig. 6.15. The unit illustrated is of the self-cleaning type, the traveling screen alternately collecting dirt from the air stream and washing it off in the fluid bath at the bottom of the circuit. Ultimately, the fluid becomes too foul for further use, and then it is drained away and a fresh charge is used.

Water performs an acceptable filtering function for many industrial conditions. The *spray washer* shown in Fig. 10.7 not only washes out many air-borne impurities but it also humidifies the air passing through, performs a cooling service, and removes certain odors.

However, the humidification of the air constitutes an undesirable act as often as otherwise. The operating cost involved in pumping water, repairing rusted areas, and replacing nozzles and piping that clog up is another detriment to a wide adoption of this type of filter. On the other hand, the cooling function may be extremely desirable in many industries. Each problem must be evaluated both pro and con, not only for a water washer, but for any type of filter.

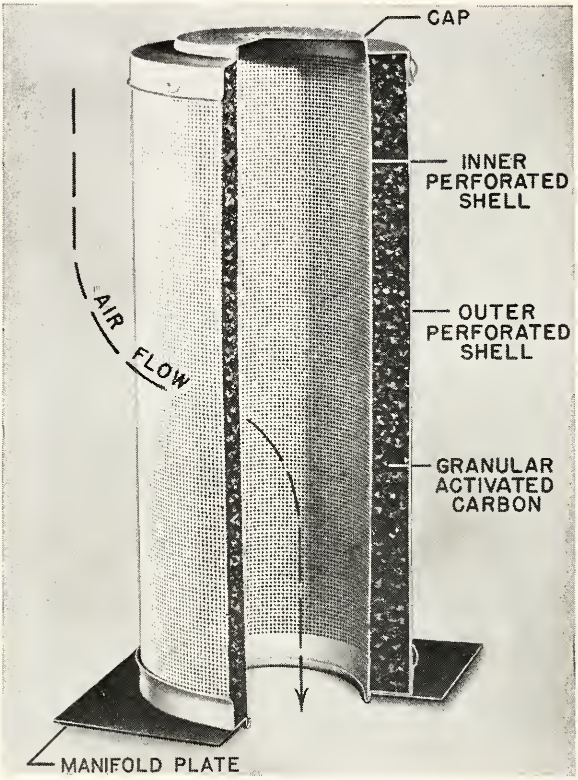


Fig. 10.8. Carbon canister, showing general arrangement and air flow. (*Courtesy of W. B. Connor Engineering Corp.*)

The *electrostatic* filter was discussed in Chap. 6 with the cinder catchers. Originally accepted in that service, its use is spreading for general air-cleaning duties. In many industries where sensitive and delicate finishing operations cannot tolerate the deposit of foreign materials, electrostatic units are regarded as standard equipment.

Odor Problems

Activated Carbon. As an all-round destroyer of odors, activated carbon is currently our most widely used agent. This material is available in canisters, as pictured in Fig. 10.8, each canister containing approximately 1 lb of carbon.

Odor-removing units are formed by framing as many canisters as are needed into a battery, the air-handling capacity of a single canister ranging from 20 to 35 cfm. In physical dimensions, a typical canister is 12 in. tall and 4 in. outside diameter.

As time passes, activated carbon becomes vitiated in service, and after absorbing from 15 to 20 per cent of its own weight in odors must be reactivated. The manufacturer accomplishes this chore for us by a reburning process; meanwhile we maintain the battery in service with replacement canisters.

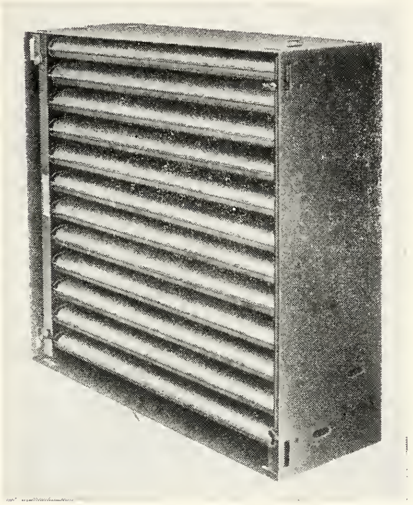


Fig. 10.9. Carbon panel for duct installation, front view. (*Courtesy of W. B. Connor Engineering Corporation.*)

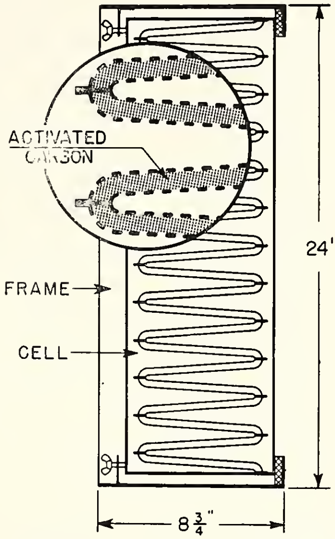


Fig. 10.10. Carbon panel, cutaway view, showing construction.

Figures 10.9 and 10.10 show an odor-absorbing arrangement particularly designed for duct installation in conjunction with unit heaters. Here the activated carbon is contained in tubes rather than canisters, and for residential installations, 2 years, service may be expected before reactivation becomes necessary.

Standard tube panels are built in depths of one, two, and three rows, and are from 16 to 24 in. wide, 20 to 24 in. long. Pressure loss should be kept to 0.14 in. W.G. for panels one tube deep; 0.16 in. for two rows deep; 0.18 in. for three rows deep. On this basis, the collection efficiency will average 20, 33, and 45 per cent, respectively.

The canister arrangement is more efficient, and will retain up to 95 per cent of the odors that enter the battery. It may be used also for outside air, whereas the manufacturer recommends the panel installation for recirculating systems only.

The Right Filter to Use

In selecting a filter, Table 10.2 provides information on which a sound selection may be based. In this table, the characteristics represent opinions in conventional practice. All filtering is relative, and although the dry-mat and viscous designs are reported as failing to remove fly ash, a critical test undoubtedly would show both of these filters to have some value in handling such material. However, other filters perform this duty more efficiently, and so for all practical purposes the chart should be followed.

TABLE 10.2. GUIDE TO FILTER SELECTION

Type of filter	Removes dust	Removes odors	Removes fumes	Removes smoke	Removes fly ash
Dry mat	Yes	No	No	No	No
Viscous surface	Yes	No	Some	Some	No
Water washers	Yes	Some	Some	No	Yes
Electrostatic	Yes	No	Some	Some	Yes
Activated carbon	No	Yes	Yes	Yes	No

The word "some" in the chart indicates that the filter will remove certain substances in the category concerned, but not all. We should clear the filter's ability in these cases with the manufacturer.

Questions

1. State the three characteristic laws of fan performance.
2. Work out a problem that demonstrates each law.
3. Describe the four basic types of fans. On what sort of service is each generally used?
4. Refer to a fan catalogue, and select a fan for 10,000 cfm against $\frac{3}{8}$-in. water pressure. What is the rpm of the fan? What is the air horsepower? What is the motor horsepower?
5. What would you do if the air to be handled is corrosive?
6. Describe the four basic types of filters.
7. Describe the two means for removing odors.
8. If you face a problem that requires the removal of both dust and odors, what would you do? Why?

CHAPTER 11

SPACE HEATERS, DOMESTIC AND INDUSTRIAL

General Characteristics

The outstanding feature of the modern space heater is its independence of ducts. The next fact worthy of note is that space heaters currently are doing more heating chores than any other single form of heating apparatus.

Space heaters as a class require less capital investment from their owners than the more elaborate means of obtaining winter comfort, which contributes mightily to their wide acceptance. Their handsomely finished cabinets are common sights in suburban and country homes, having been purchased in many instances from mail-order catalogues.

In addition to their low first cost, residential space heaters when fired by either gas or oil possess another advantage: practically no attention is required except a seasonal cleaning. Coal-fired units of course demand periodic refueling and the removal of ashes, but as with the others maintenance is negligible.

The cheapest design of space heater, which is also, in consequence, the most numerous, operates on the principle of simple convection. Cool air enters the bottom of the unit and, being swiftly warmed, rises to the top where it emerges into the room.

Theoretically, the heater-warmed air fans out across the ceiling until it meets the walls where its growing coolness causes it to sink to the floor and return to the bottom of the heater. Actually, it does just this, but the movement is so slow that a wide difference in air temperatures between the ceiling and floor is both common and annoying. In consequence, space heaters occasionally are criticized adversely by persons who do not consider hot heads and cold feet natural to human comfort.

By spending a little more money, we can obtain space heaters in which the air circulation is expedited by a fan. As a result, the 85 F ceiling and 60 F floor common to the natural convection units assume the much better relationship of 80 F and 70 F. However, in mild climates, the small extra cost of the fan is enough to keep it from enjoying unanimous favor.

In the industrial applications of space heaters, a fan always is part of the unit. In fact, we will encounter large heaters in which, not one, but several fans are busily employed in discharging warm air. Multiple-fan designs generally use a single large motor driving the fans through a common shaft.

Since industrial heaters frequently must distribute the warm air over large areas and to all points of the compass in locations such as warehouses and factories, discharge hoods, or cowls, are frequent adjuncts to the discharge openings. These cowls are movable both up and down and around the horizon, and by pointing their mouths toward the areas we wish to heat, the warm blasts are delivered effectively.

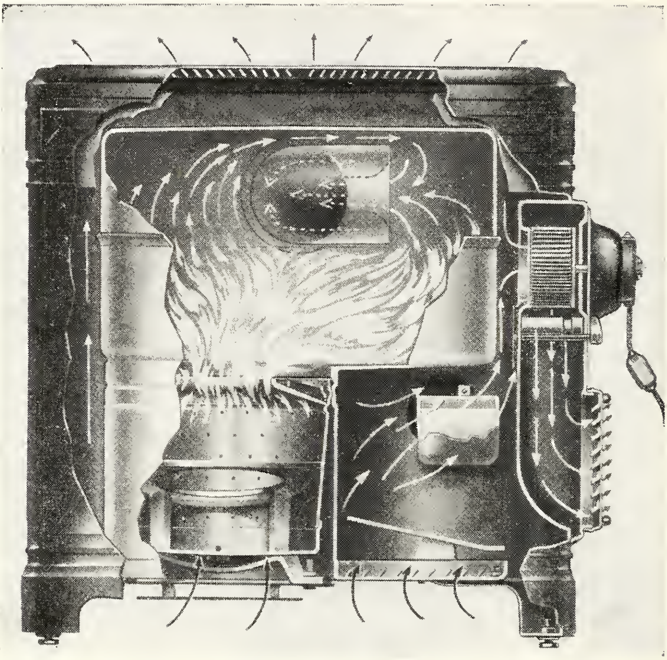


Fig. 11.1. Cutaway view of space heater, showing blower for discharging air to room, humidifier, and the dual-chamber pot burner. *(Courtesy of Duotherm Division of Motor Wheel Corporation.)*

Direct-fired Residential Units

Sizes and Capacities. The residential units, frequently called "circulators," range in capacity from 20,000 Btu per hr up to 75,000 Btu per hr. On volume ratings, a 20,000 Btu unit is recommended for 2,000 cu ft of average, or everyday, construction with an outside temperature of

0 F. Under the same conditions, a 40,000 Btu output will serve 4,000 cu ft, a 53,000 Btu output, 5,300 cu ft, and the largest, 75,000 Btu, is recommended for 7,500 cu ft of house volume.

All the oil-fired heaters employ pot-type burners, with an integral oil tank sized for from 2 to 7 gal. Where physical conditions permit, many owners feed fresh oil by gravity from an elevated main tank located outdoors. This arrangement is not favored by fire departments, since the occurrence of a fire can impose difficulties in stopping the oil feed.

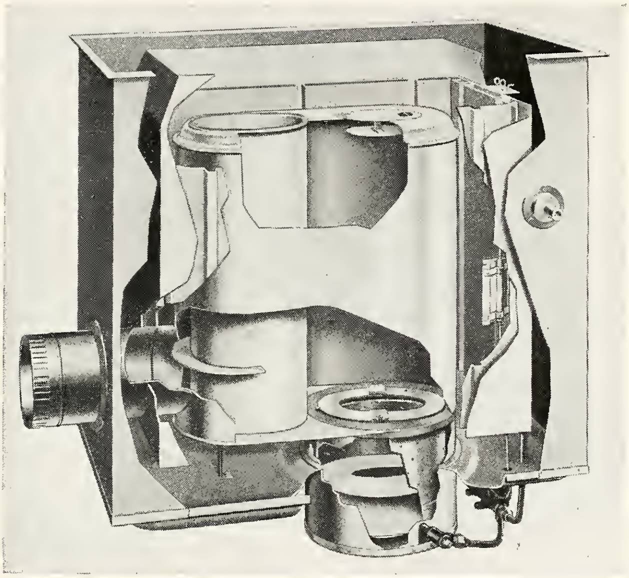


Fig. 11.2. Cutaway view of floor furnace, oil-fired, showing arrangement of pot-type burner, vertical hand control levers at right, and smoke outlet at bottom left. (*Courtesy of the Coleman Company, Inc.*)

In styling, one manufacturer offers cabinet finishes in both the Hepplewhite and Chippendale tradition, with a cumulative effect not dissimilar to that of a fine radio.

Figure 11.1 is a cutaway view of the space heater's internal arrangement. These units may be controlled by a room thermostat, if desired. Built-in humidifier pans are standard equipment.

Floor Furnaces

Features of Design. In order to conserve floor area, the floor-furnace type of space heater has been developed for small homes. Figure 11.2 illustrates a conventional design, in which the air circulation results from convection.

Air-moving fans, to speed up the delivery of heat, are provided with some units at an extra charge.

Tests at the National Bureau of Standards indicate that heat distribution from a floor furnace operating on convection is somewhat better than from a space heater operating on the same principle. Apparently the improvement results from the lower average elevation of the return air to the floor heater.

Floor heaters are not recommended for structures in which the warm air is required to pass through more than one doorway. For this reason, rambling houses frequently will be observed with two or more small units located throughout the building.

Floor heaters are available in sizes from 25,000 Btu to 75,000 Btu per hr output. Breeching connections to the chimney should never exceed 10 ft, and an 8-ft limit is even better. The breeching should be insulated with at least $\frac{1}{2}$ in. of asbestos in mild climates, 1 in. in cold climates.

A special type of floor heater is built to supply heat to both sides of a partition. This design may employ a simple floor grille, common to both areas, or use a vertical stack that rises for some inches into the partition, the warm air emerging from registers or grilles set flush with the walls.

Floor furnaces may be obtained for either gas or oil firing, the latter design employing gravity feed to a pot-type burner.

Due to their mounting beneath the floor, lint and dust accumulate in the air passages. These may be cleaned with a damp cloth wrapped around a broom or, better, with a vacuum cleaner. Neglect to remove these accumulations produces a fire hazard during times of heavy demands for heat, the lint igniting readily from an overheated burner.

Electric Heaters

Radiant Panel Heaters. Figure 11.3 shows a design of glass panel that is being used successfully in areas of moderate- to low-cost electricity.

The panel is constructed of an aluminum grid which is fused into a plate of tempered glass, approximately $\frac{1}{4}$ in. thick. The glass is mounted in ceramics and is separated from its aluminum reflecter by an air space. The standard size of the steel containing frame is 21 by 31 in. Each panel is rated at 1,000 kw, and may be used with either alternating or direct current. Additional data on this unit are given in Chap. 9.

Electric Air Heaters. The familiar electric heater that we carry around from room to room employs an electric heating element over which the air moves by convection or by a fan if we are willing to pay a little more.

Some electric heaters are designed for installation in walls; common sizes are given in Table 11.1.

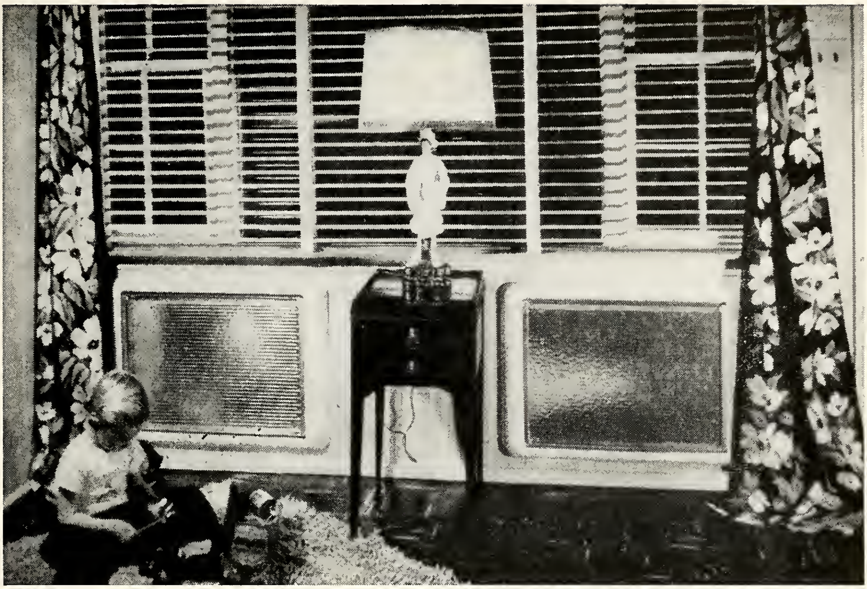


Fig. 11.3. Radiant glass panels recessed in wall under window. *(Courtesy of Continental Radiant Glass Heating Corporation.)*

Large units for factory or warehouse heating may be obtained, using 60 kw per hr and having a rated output of over 200,000 Btu. With an air temperature rise of 65 deg, the air handled amounts to 4,000 cfm.

Table 11.1. Sizes of Electric Heaters Designed for Installation in Walls

Kw	Max output, Btu/hr	Air handled, cfm	Air temp. F
2	6,824	110	125
3	10,236	110	155
4	13,648	170	140
5	17,060	170	160

Direct-fired Industrial Heaters

Sizes and Capacities. Figure 11.4 shows a style of heater commercially available in sizes ranging from 400,000 Btu per hr up to 2,000,000 Btu per hr. In somewhat varied forms, we can obtain smaller sizes, down to 65,000 Btu per hr. These industrial units are built for all types of firing, oil, gas, and coal. Table 11.2 presents useful data on their physical dimensions and capacities. Temperature rise through the heater is conventionally rated at 80 to 84 deg.

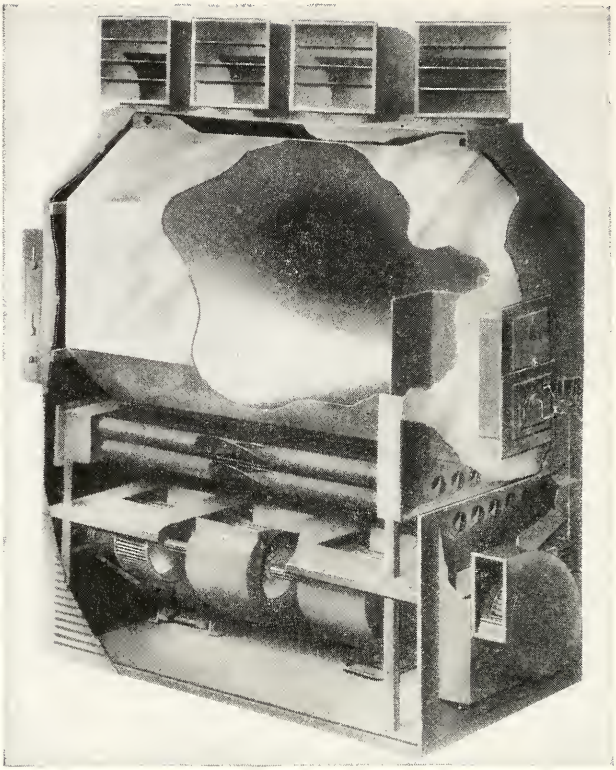


Fig. 11.4. Cutaway view of large industrial space heater with discharge cowls. (*Courtesy of the Dravo Corporation.*)

Filters are not standard equipment with the large heaters, primarily because of the great volume of air being passed through the machine. The filter area required is simply too great to be included in the cabinet, and if filtering is desired, additional facilities must be provided.

Table 11.2. Physical Dimensions and Capacities of Heaters

Output, Btu/hr	Air output, cfm (referred to 70 F)	Hp of fan motor	Dimensions		
			Width	Length	Height
400,000	4,500	1½	2′7″	4′11″	8′1″
1,000,000	11,000	5	3′8″	7′3″	10′0″
1,500,000	17,000	10	4′3″	8′11″	11′2″
2,000,000	22,000	20	4′9″	9′5″	12′9″

Small heaters, normally handling up to 3,000 cfm, usually are fitted by their builders with filter mats, mostly of the dry-mat type.

Air Velocity. When we locate space heaters in an industrial area, some care must be exercised to avoid the imposition of drafts or hot blasts on individuals working in the area. Various researchers have investigated the effects of air movement on the human skin, and report the following data, particularly from Peele:

Velocity of Air, Fpm	*Sensation in the Individual*
60	Barely perceptible
180	Pleasant movement
300	Unhealthful, if perspiring

Questions

1. What are the outstanding virtues of space heaters?
2. What are the limitations of space heaters?
3. What size of space heater would you require for your own home? Calculate the heat losses as directed in Chap. 2.
4. Describe the characteristics of the floor furnaces.
5. Where would you use a panel-type electric heater? Why?
6. Describe the outstanding features of the industrial space heaters.

CHAPTER 12

DISTRICT STEAM HEATING

Rate Considerations

When we contemplate the purchase of steam for heating and perhaps process work, we face at once the economic problem of cost. Will purchased steam cost more, or less, than the operation of our own boiler plant?

Conversely, if we are in the seller's position, a common one in industrial areas, what price should we place on the steam that we dispatch to serve our neighbors?

In either case, we must consider the question of rates in some detail.

The Space, or Radiation, Charge. Pioneering district steam companies sometimes assessed their customers on the basis of building volume or connected radiation. For example, one Illinois company charged 8 mils per year per cubic foot of building—with special consideration given to parlors and garages. On the connected radiation basis, charges could be expected anywhere between 19 cents per square foot per season and 70 cents.

These methods of charging are still being practiced, occasionally quoted as a base charge, with an added assessment for actual steam consumption, as metered.

In the opinion of this writer, any charge based wholly on volume or connected radiation is likely to lead straight to dissatisfaction for all concerned. The purchaser often feels he is paying for something he fails to get, especially if much of the building is unused; the seller, if he finds himself losing money on the contract, may accuse the buyer of wasting steam, and attempt to raise the rate. Steam is not cheap; it cannot be generated cheaply with fuel and labor charges as they are; consequently when dealing with this expensive product, we should meter all we sell and bill accordingly.

Consumption Metering. We have two means for measuring steam consumption. The first and cheaper practice is to use a condensation meter, as illustrated in Fig. 12.1. This meter actually weighs the water prior to its return to the boiler feed system, or to its dumping in the sewer.

Condensation meters are available in standard sizes from 150 to 15,000 lb per hr. If the load exceeds the capacity of a standard meter, we simply pipe up the condensate to as many meters as we need.

Flowmeters are illustrated by Figs. 12.2 and 12.3. These instruments are obtainable in any size and capacity.

The Shuntflo type is connected directly into the steam line, with some of the steam by-passed through a registering gear train, as shown. This meter is a totalizing affair, always showing the total passage of steam up to the moment of reading.

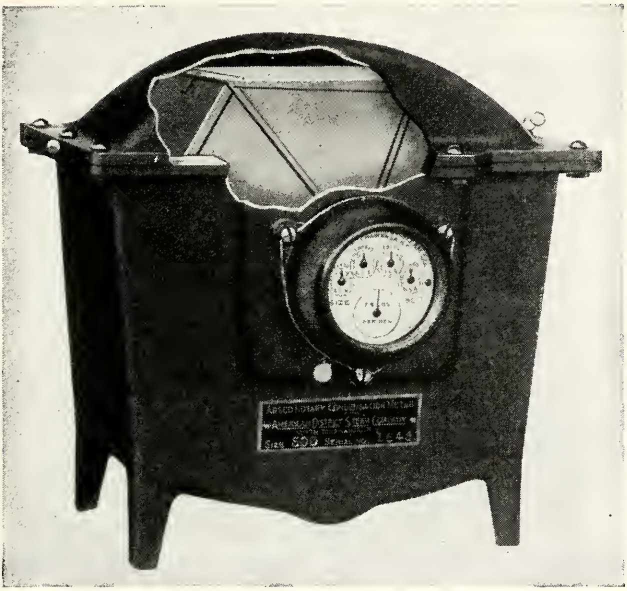


Fig. 12.1. Condensate meter. (*Courtesy of American District Steam Company.*)

The flowmeter shown in Fig. 12.3 employs the familiar orifice in the steam line principle, with pressure piping connecting each side of the orifice to the meter. This meter shows the flow of the moment, and, additionally, records variations on a 24-hr chart, as desired. The actuating force is the differential pressure over the orifice.

This type of meter is likely to cost more than either of the others, but it is a favorite with district steam companies because of the recording feature. If steam is being used abnormally, the story is mutely presented by the chart. Figure 12.4 shows the internal workings of this meter.

Consumption charges generally are based by the district steam companies on the actual cost of generation. This cost incorporates all the fixed capital charges on the boiler plant, depreciation, insurance, interest, maintenance, etc., together with labor, electricity, water, fuel, and ash handling.

The consumption charge frequently includes, as it should, a factor to compensate for fluctuations in the price of fuel at the plant. One district steam company states this situation substantially as follows:

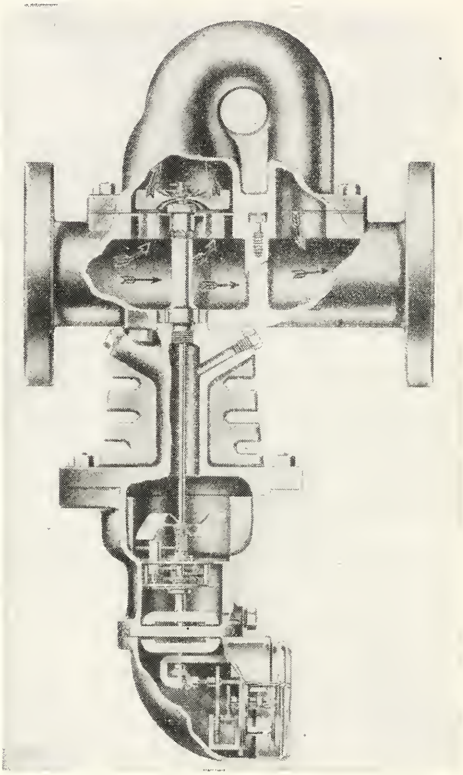


Fig. 12.2. Shuntflow type of steam flowmeter, showing internal arrangement of instrument. (*Courtesy of Builders-Providence, Inc.*)

"The consumption charge is based on a charge for coal delivered to the plant of $4.35 per net ton. When the average cost of coal so delivered exceeds $4.50 per ton, or drops to less than $4.20 per ton, the consumption charge shall be adjusted at the rate of 3 mils per 1,000 pounds of steam, for each 5 cent increase or decrease above the basic average ton cost of $4.35. The average cost and adjustments are made for 3-month periods at a time."

If the consumption charge is the only charge, it should include also an item of profit for the seller.

Consumption charges sampled throughout the country reflect local conditions by the observed variation of rates, which fluctuate all the way from 50 cents or less per 1,000 pounds, up to $2. Many companies employ a block rate, that is, the first few thousand pounds purchased in a month are at a high rate, with succeeding consumption billed at a lower figure. For example, one company quotes:

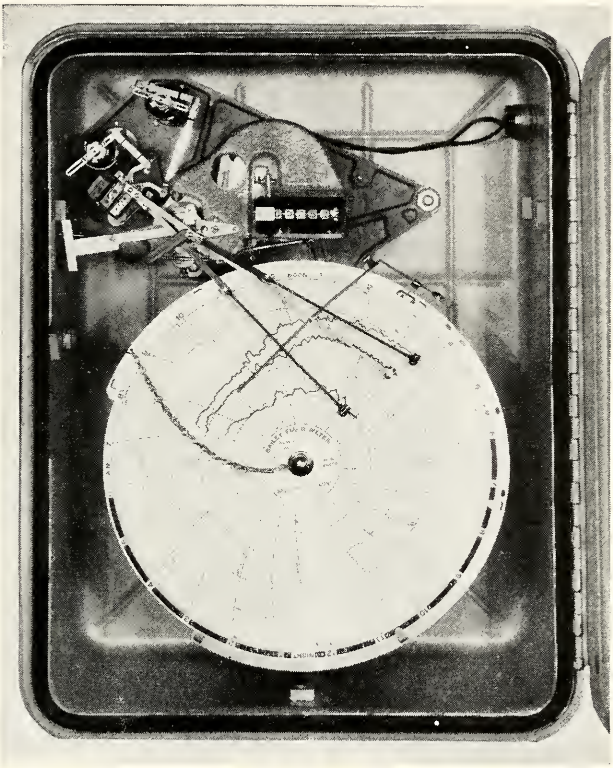


Fig. 12.3. Pipe-orifice type of steam flowmeter, with recording chart. (*Courtesy of Bailey Meter Company.*)

First 25,000 lb per month	$1.05 per thousand
Next 25,000 lb per month	0.90 per thousand
Next 50,000 lb per month	0.75 per thousand
Next 50,000 lb per month	0.68 per thousand
Balance at	0.60 per thousand

The economics of the block rate is based on the premise that once service is established it is possible to sell additional steam at the increment cost of material and labor, provided it is not necessary to supply additional plant or facilities.

Demand Metering. The practice of demand metering is intended to assess each customer with his just proportion of the plant investment that is needed to meet his requirement. For example, a customer who has a peak demand of 3,450 lb of steam per hour actually possesses an option on approximately 100 boiler hp in the steam generating plant at the time of the peak. Perhaps he uses this peak only once a day, but the boiler capacity must be ready when he calls on it.

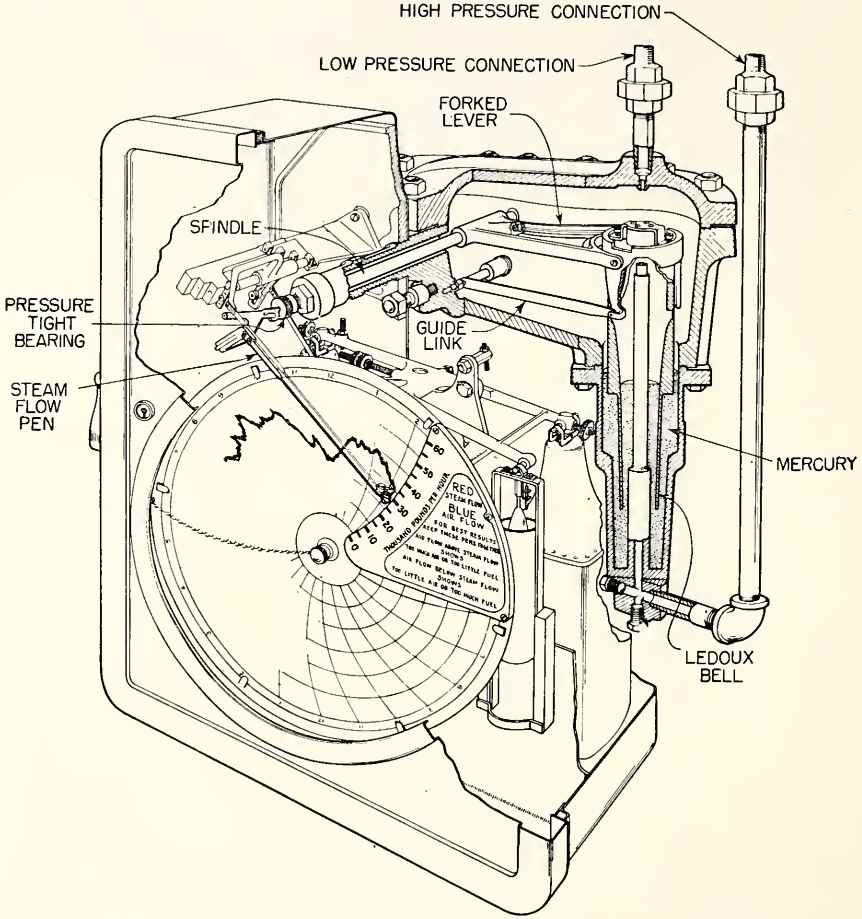


Fig. 12.4. Orifice type of meter adapted to boiler control. Note internal arrangement of parts and facilities for recording both steam and combustion air flow. (*Courtesy of Bailey Meter Company.*)

Steam demands customarily are measured on an hourly basis; the demand for each hour the steam is turned on is registered on a chart. The billing peak for any given season is the average of the three highest peaks recorded in the maximum month. Thus if the month of January shows peaks of 1,200 lb per hr, 1,600 lb, and 1,800 lb, the average peak becomes 1,534 lb per hr.

The rate for the demand varies, large companies assessing it on the basis of 12 to 15 cents per pound per month, with the customer responsible for meeting the charge every month during the heating season. At a rate of 12 cents, the seasonal charge for an 8-month season on a 1,534 lb peak would be $1,472.64. This demand billing is sometimes called a ready-to-serve charge, or a capacity charge, the metered consumption of condensate being additional.

Demand charges are not often instituted by industrial plants undertaking to supply steam to a neighbor, since the steam plant usually is already in existence designed for the seller's requirements. In these cases, an increment cost charge plus a profit is conventional practice. Who pays for the physical extension of piping is then settled by negotiation.

Steam Distribution Problems

How Much Should the Steam Main Cost? Public utility companies use a rule of thumb in considering service extensions which is expressed this way: The total gross revenue from a service extension should equal the physical cost of the extension in from 2½ to 3 years. In short, if a prospective customer will pay $2,000 a year for his steam service, the seller is justified economically in spending from $5,000 to $6,000 to build the line.

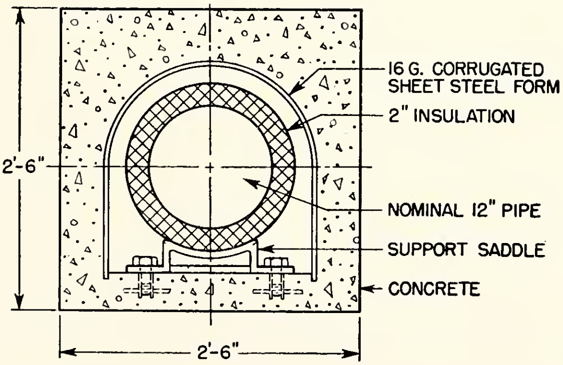


Fig. 12.5. Characteristic arrangement of concrete duct for underground steam mains.

Construction costs vary with the city area in which the line is to go, and with the cities themselves. A design of underground main as shown in Fig. 12.5 may be expected to cost under a city street $50 per foot for a 6-in. pipe. Larger pipes will cost more in proportion.

Types of Underground Construction. The design shown in Fig. 12.5 is common where heavy street traffic exists. The sheet-metal form is left in place when the roof is poured.

The wood-stave conduit design is illustrated by Fig. 12.6. While wood is generally thought to possess very doubtful stamina when buried in the soil, conduits of this sort have a surprisingly long life. Apparently they derive some protection from the heat of the steam pipe, percolated moisture evaporating faster in the warm earth than it would if the structure were cold.

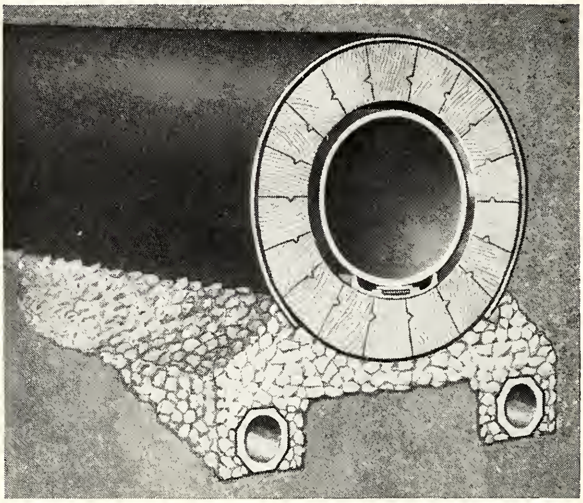


Fig. 12.6. Wood-stave conduit as manufactured by American District Steam Company.

Probably the form of conduit most often employed is formed of precast tile; this is illustrated in Fig. 12.7. This design is both strong and tight.

Returning of Condensate. Salvaging of the condensate for return to the boiler is primarily a matter of economics. For long piping runs, we have problems of pumps, larger conduits, and more rollers and supports. Many of the large district steam companies find disposal of the condensate on the customers premises to be cheaper than its recovery. Before making a decision, comparative estimates are necessary; on one side there is the physical cost of the return, and on the other, there is the expense of buying make-up water for the boilers and of treating it after it is bought.

Pressure and Temperature. Steam distributed at high pressures offers a substantial saving in pipe sizes, but involves some manner of pressure reduction in most cases at the delivery end.

The effect of pressure on pipe size may be readily visualized if we note that, at 10 psig, a pound of steam occupies 16.3 cu ft, while at 150 psig, its volume shrinks to 2.752 cu ft per lb. Specifically, if we have 10,000 lb of steam to convey, the lower pressure will require an 8-in. pipe, the higher pressure a 4-in. pipe.

Reducing valves at the delivery end and relief valves to protect the customers' low-pressure equipment may be offset somewhat in cost by sizing the steam pipe so stringently that most of the desired pressure drop is produced by pipe friction. Any of the standard pressure-drop formulas may be used to determine the pipe size, although the following is satisfactory:

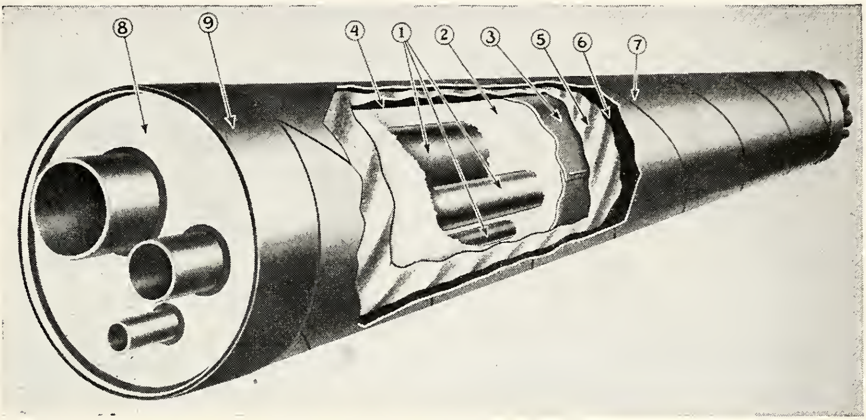


Fig. 12.7. Preformed tile conduit for underground piping. 1. Fluids-carrying pipe. 2. Insulation. 3. Asbestos-felt jacket. Asphalt saturated if necessary. 4. Air space. 5. 16-gauge corrugated, galvanized, welded conduit, inside coated with baked-on phenolic resin. 6. Asphalt coating. 7. Asbestos-felt. Asphalt saturated. 8. Preseal ring. 9. Smooth end, iron collar. (*Courtesy of the Ric-Wil Company.*)

$$P = \left(\frac{fW^2V}{d^5}\right) 0.000336$$

where P = pressure drop per 100 ft of pipe, psig

f = friction factor (this is a variable being computed by means of the Reynolds number equation; for approximations, we may use a factor of 0.016 for clean steel pipe in sizes between 1 and 12 in., since the flow is turbulent in most cases)

W = steam discharge, lb per hr

V = specific volume, cu ft per lb

d = internal diameter of the pipe, in.

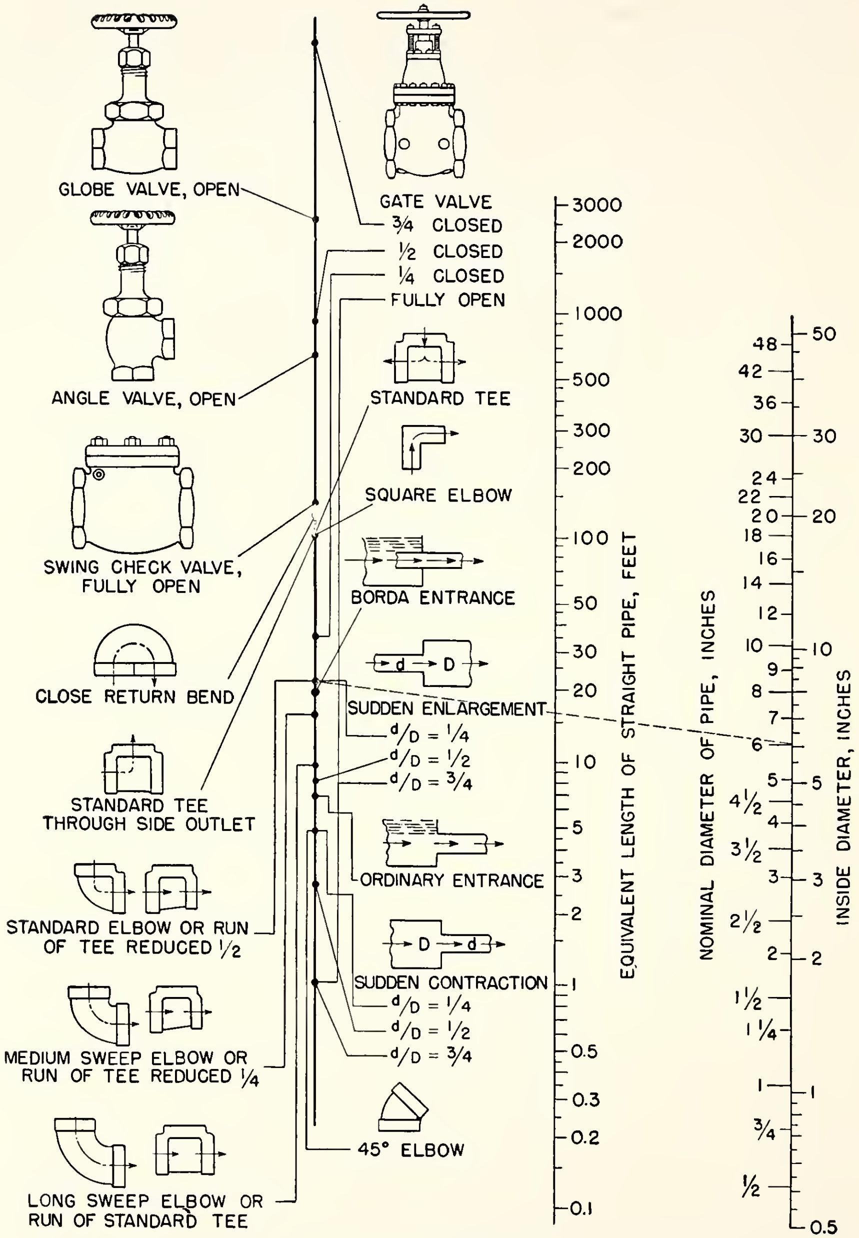


Fig. 12.8. Chart of pressure drop through valves and fittings. (Courtesy of Crane Co. from *Technical Paper No. 409.*)

The pressure drop resulting from the flow of fluids through valves and fittings may be taken from the chart (Fig. 12.8). Drops through the various types of pressure-reducing valves should be obtained from the manufacturer's catalogue.

High temperature for steam in transit, obtained by superheating the steam, performs a valuable service in checking condensation losses in the main. This benefit is the result of the superheated steam's failure to wet the inner surface of the pipe, a film of moisture, such as accrues from saturated steam, being an active agent in the transfer of heat from the vapor to the iron.

For estimating, we may expect superheating to cut line condensation losses by 40 per cent, if compared with saturated steam losses at comparable pressures. For exact data, we should compute the losses, of course, across the insulation thickness and temperature difference being used.

For the same reason that superheat is an asset to steam in transit, its presence is undesirable in radiators. There its reluctance to condense causes a substantial reduction in the heat-emitting capacity of the radiator, since most of steam's heating value resides in the release of its latent heat.

If high superheat is present in an unsatisfactory installation, we should give immediate thought to desuperheating. Correction is possible by the use of a commercial desuperheater or, alternatively, by a homemade water spray introduced into the steam pipe. The latter expedient is not accurate, sometimes induces water problems in the steam, and should be undertaken only when the cost of commercial treatment cannot be accepted.

Occasionally, superheat in the steam main causes faulty operation of the bucket traps which serve the main. The trouble shows up in a tendency of the traps to keep on blowing after emptied of water. This condition can be cured by removing the insulation from a foot or two of the trap inlet piping or, alternatively, by increasing the size of the water leg formed by this piping. In extreme conditions, we may need to exercise both treatments.

Manholes and Service Connections. A typical manhole is illustrated by Fig. 12.9. Expansion joints are located in these manholes not only for accessibility, but to serve as points for originating service branches to customers along the route. The center of the standard slip joint is anchored, and above the anchor a pipe nozzle is provided to which the branch may be attached. Figures 12.10 and 12.11 illustrate popular forms of expansion joints.

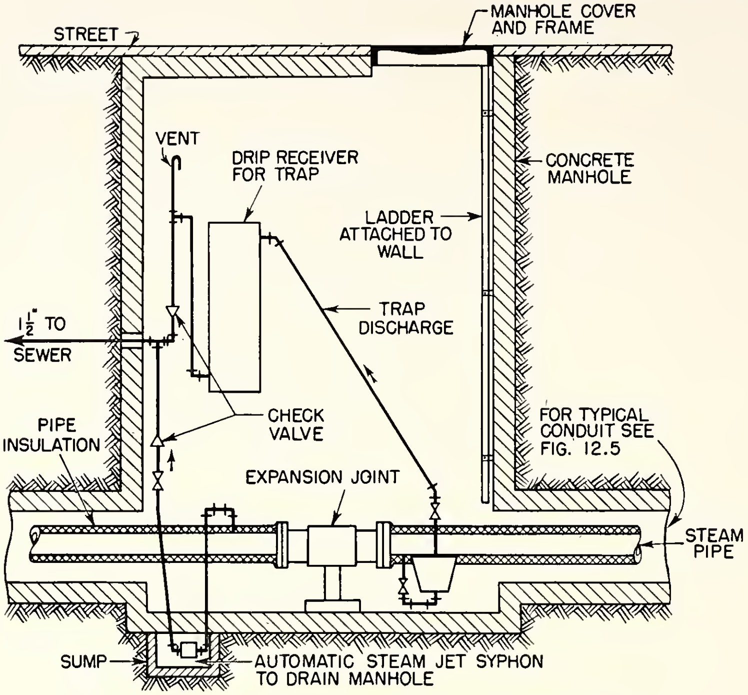


Fig. 12.9. Manhole arrangement for city construction.

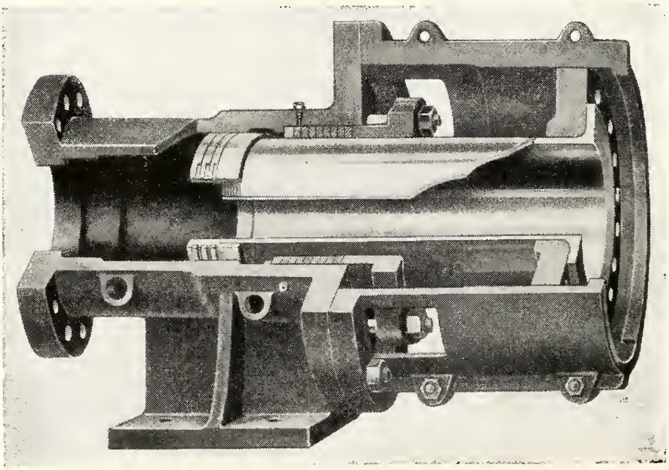


Fig. 12.10. Slip type of expansion joint. (Courtesy of American District Steam Company.)

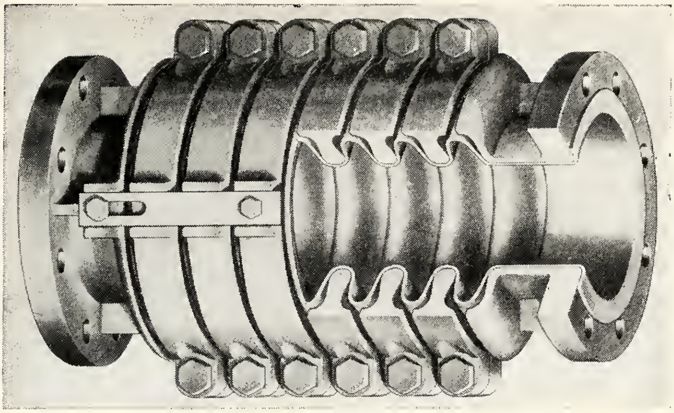


Fig. 12.11. Packless type of expansion joint. (*Courtesy of American District Steam Company.*)

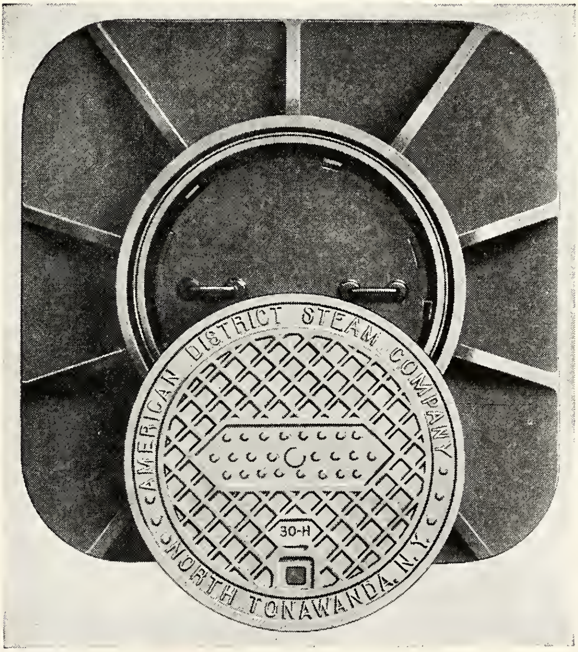


Fig. 12.12. Manhole cover, suitable for heavy traffic locations. (*Courtesy of American District Steam Company.*)

Alternatively, we may design the line with expansion bends, still taking off service at the points of anchorage.

Figure 12.12 shows a typical manhole cover.

Overhead Distribution Mains. Overhead distribution mains are not suitable for city conditions, but serve very well in conveying steam from one plant to its adjacent neighbors, or to distant parts of its own property.

Overhead mains eliminate manholes and simplify pumping problems. On the other hand, we must design carrying towers, single pole for one or two pipe lines if the pipe is small, A-frame towers for large and multiple pipe systems. There is also the problem of weather protection from direct impact of the elements, and the need to safeguard the various lines from freezing at times of low load. Against the elements, we may place on top of the insulation either sheet-metal covering or an asphalt roofing with overlapping joints securely wired on. For freezing, adequate drainage is the best protection, plus of course plenty of insulation.

Rollers for supporting the pipe and either expansion joints or expansion bends are essential.

Underground piping is more sightly, but overhead piping offers possibilities for a saving on the investment.

Entering the Customer's Premises

The Connection Design. Since most district steam systems operate at rather high pressures, say from 50 psig up to 200 psig, we must provide pressure-reducing stations somewhere ahead of the building's heating system.

Figure 12.13 illustrates an arrangement for handling the reduction in two steps. There is a definite advantage in this layout, since the intermediate pressure permits the installation of a flow-limiting orifice under consistently uniform conditions of pressure. This means the customer can vary his own system pressure as he wishes by adjusting the final reducing valve, without disturbing the maximum flow through the orifice.

Some district steam companies use the orifice design for setting up a demand charge, the charge being based on the maximum capacity of the orifice in pounds per hour.

Even if we have no orifice in our plans, the use of two reducing valves is often desirable for low-pressure distribution, and of course is essential for high-pressure systems. Most heating systems operate normally on a pressure of from 1 to 2 psig (pounds per square inch gauge), and if the incoming service main carries 150 to 200 psig, the standard pressure reducing valves cannot accommodate so large a differential in a single step.

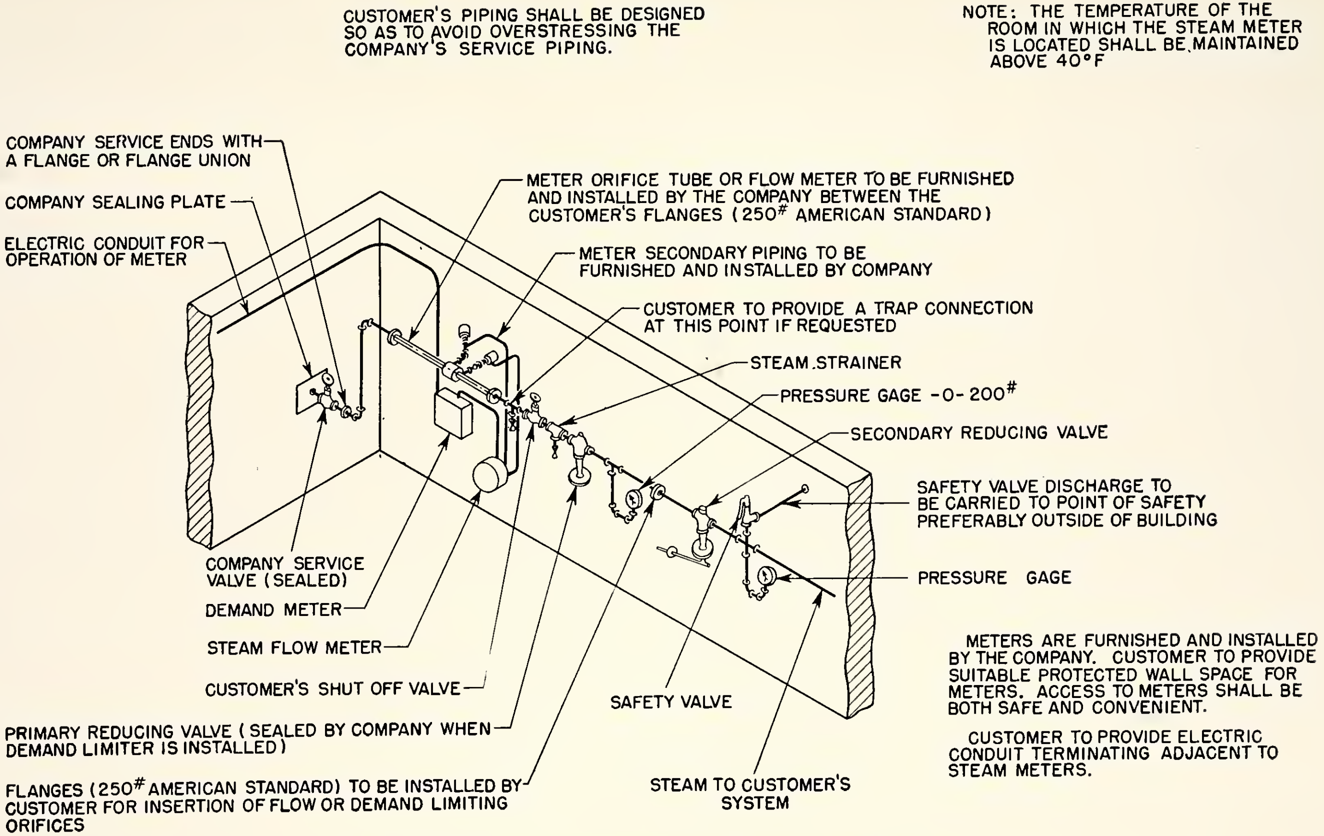

CUSTOMER'S PIPING SHALL BE DESIGNED SO AS TO AVOID OVERSTRESSING THE COMPANY'S SERVICE PIPING.
NOTE: THE TEMPERATURE OF THE ROOM IN WHICH THE STEAM METER IS LOCATED SHALL BE MAINTAINED ABOVE 40°F
COMPANY SERVICE ENDS WITH A FLANGE OR FLANGE UNION
COMPANY SEALING PLATE
ELECTRIC CONDUIT FOR OPERATION OF METER
METER ORIFICE TUBE OR FLOW METER TO BE FURNISHED AND INSTALLED BY THE COMPANY BETWEEN THE CUSTOMER'S FLANGES (250# AMERICAN STANDARD)
METER SECONDARY PIPING TO BE FURNISHED AND INSTALLED BY COMPANY
CUSTOMER TO PROVIDE A TRAP CONNECTION AT THIS POINT IF REQUESTED
STEAM STRAINER
PRESSURE GAGE -0- 200#
SECONDARY REDUCING VALVE
SAFETY VALVE DISCHARGE TO BE CARRIED TO POINT OF SAFETY PREFERABLY OUTSIDE OF BUILDING
PRESSURE GAGE
COMPANY SERVICE VALVE (SEALED)
DEMAND METER
STEAM FLOW METER
CUSTOMER'S SHUT OFF VALVE
SAFETY VALVE
PRIMARY REDUCING VALVE (SEALED BY COMPANY WHEN DEMAND LIMITER IS INSTALLED)
FLANGES (250# AMERICAN STANDARD) TO BE INSTALLED BY CUSTOMER FOR INSERTION OF FLOW OR DEMAND LIMITING ORIFICES
STEAM TO CUSTOMER'S SYSTEM
METERS ARE FURNISHED AND INSTALLED BY THE COMPANY. CUSTOMER TO PROVIDE SUITABLE PROTECTED WALL SPACE FOR METERS. ACCESS TO METERS SHALL BE BOTH SAFE AND CONVENIENT.
CUSTOMER TO PROVIDE ELECTRIC CONDUIT TERMINATING ADJACENT TO STEAM METERS.

Fig. 12.13. Typical service connection on customer's premises.

Control Methods. To be economical, district steam requires some form of automatic control. The simplest arrangement is seen in the familiar thermostat, which is readily set up to actuate a valve in the steam line. Frequently this sort of control does its job by moving the lever of a weight-loaded reducing valve up and down as may be necessary to increase or decrease the input of steam.

The same thermostat can be used to operate a steam line valve between fully open and fully closed positions, without any attempt at modulation. When thus disposed, no steam flows unless the room temperature falls below the preset temperature.

The fact that a steam radiator literally may be packed with steam to condense later at leisure has produced a type of time-clock control, specifically designed for operation on demand charge service. Its function is to admit steam into the building system at the full capacity of the service pipe for a predetermined period, say 15 or 30 min, then interrupt the input for a period of equal length. The net result is a high instantaneous pull on the district steam mains, but a comparatively low demand charge against the customer, with a better than normal distribution of steam to distant radiators.

A system that is old and contains many areas of high resistance can be made to function satisfactorily occasionally by this plan of steam flooding.

Another approach to the control problem is to insert orifices in radiator inlets. This method is discussed in detail in Chap. 3.

For steam line valves, we have our choice of diaphragm-actuated valves, spring-loaded valves, weight-loaded valves, or solenoid-operated valves. The cheapest of these designs is the weight and lever type, with the weight counterbalanced by a diaphragm. These valves have a single seat, a desirable feature where complete shutoff is concerned, since a single seat valve is easier to keep tight. In operation, the thermostat moves the lever up and down as may be required to control the steam.

Questions

1. How is the space charge or connected radiation charge computed?
2. Describe three methods currently in practice for consumption metering.
3. What do you think is the advantage of each method?
4. What is a fuel adjustment clause? How is it applied?
5. What is meant by a block rate?
6. What is demand metering? What is its purpose?
7. State a rule of thumb for justifying the installation of a steam main.
8. Describe three forms of underground steam conduit.

9. Why is condensate sometimes discharged to waste, rather than returned to the steam plant?

10. Is there any advantage in distributing steam at high pressure?

11. Would you superheat steam you planned to sell to a neighbor? Why?

12. How would you design a steam line to take a pressure drop from 100 to 10 psig?

13. What is the advantage of using overhead mains, if any?

14. How would you prepare an outside main against weather?

15. Describe three types of steam control.

CHAPTER 13

THE HEAT PUMP

What the Heat Pump Is

The Operating Cycle. The heat pump should be no mystery to any of us, since it is essentially as simple as a domestic refrigerator. In the refrigerator, a liquid known as a "refrigerant" vaporizes in the freezing compartment (technically known as the evaporator) with the necessary heat of vaporization being obtained from the air and foodstuffs in the refrigerator. This vapor is sucked out of the evaporator by a compressor, and is promptly transferred from the compressor to a condenser, where it loses its burden of heat and becomes liquid again.

In most of our domestic refrigerators, the cooling medium that assumes the refrigerant's heat is simply the room air, which is arranged to pass over the condenser by natural convection, and consequently the heat from inside the refrigerator eventually ends up in the room. Obviously, if our refrigerator were big enough and ran long enough, we could heat up first the room in which the refrigerator stands, and then the whole house.

Heat pumps for building heating operate in this same way. Only the source of the heat for transfer to the room air is different. In some designs the evaporator (instead of being set inside the refrigerator) is located outdoors, and the heat of vaporization is obtained from the surrounding air. There is ample heat waiting to be harvested in the great outdoors, even if the thermometer stands at zero, for the lowest temperature currently believed possible and called the absolute zero is 459.7 degrees below our familiar Fahrenheit zero. When exposed to this vast reservoir of heat, the refrigerent promptly absorbs so many Btu that it vaporizes. The vapor thus formed is then drawn out of the evaporator by the compressor, and is transferred under pressure to the condenser, where the heat is removed and used for comfort heating.

Figure 13.1 illustrates this sort of circuit, with a fan distributing the condenser-warmed air to the building through a system of ducts.

As an alternative to using outdoor air for a heat-pump installation, we can use water if we have it, or even the earth itself. Reports from Zurich, Switzerland, tell us that a part of that city is being heated with Btu extracted from the water beneath an ice-bound river. That is spectacular, but no better than our American practice of using well water, or city water from the street mains.

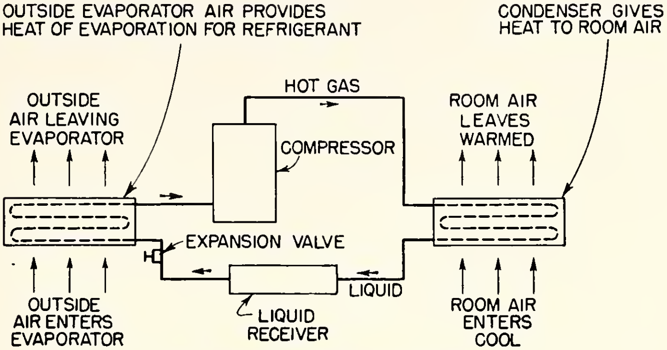

OUTSIDE EVAPORATOR AIR PROVIDES HEAT OF EVAPORATION FOR REFRIGERANT
CONDENSER GIVES HEAT TO ROOM AIR
OUTSIDE AIR LEAVING EVAPORATOR
HOT GAS
COMPRESSOR
ROOM AIR LEAVES WARMED
EXPANSION VALVE
LIQUID
LIQUID RECEIVER
OUTSIDE AIR ENTERS EVAPORATOR
ROOM AIR ENTERS COOL

Fig. 13.1. The heat-pump circuit.

Earth as a heat source is being investigated industriously by both the commercial manufacturers of heat pumps and the Armour Research Foundation of the Illinois Institute of Technology. We all know that ground temperatures remain comparatively high, from 40 to 50 F throughout the winter, provided we take the temperature below the frost line. To obtain this heat for vaporizing the refrigerant, coils or pipes are driven into the earth, as shown in Fig. 13.2.

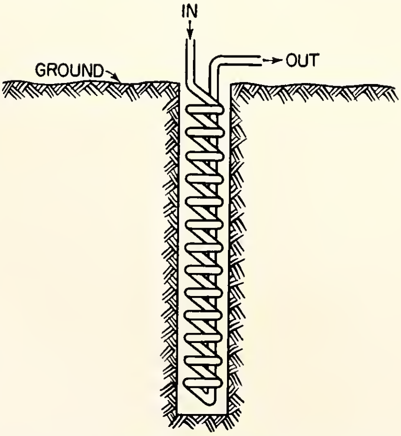


Fig. 13.2 Ground coils at St. Louis, Mo.

Some designs evaporate the refrigerant right in the ground, while others employ a primary circuit of either water or a nonfreezing liquid that picks up heat in the coils and carries it to a heat exchanger. The refrigerant then vaporizes in the heat exchanger.

By use of the exchanger, we relieve the compressor of considerable friction loss on the suction side, but the location of the exchanger can worry us too. As an illustration, one manufacturer describes his exchanger as being in the form of a rectangular tank 26 in. by 36 in. by 6 ft 0 in.

long, and there is not always space available for these far from dainty dimensions.

The heat pump has been so named as a result of the compressor's duty in the circuit. Every time we compress anything, whether it happens to be air or the vapor from a refrigerant, the act of compression produces heat. Consequently, the vapor that finally reaches the condenser actually has acquired heat from two sources: (1) from the air, or water, or earth and (2) from the squeeze that it undergoes in the compressor. In other words, we have taken a certain amount of heat and "pumped it up" to more heat by mechanical means.

Occasionally the procedure is referred to as "reversed refrigeration." This designation probably is less appropriate than the term heat pump in view of what happens; in fact with equal reason we might refer to ice making as reversed heating. However, some engineers prefer it, and no doubt it will continue to be used.

Air as a Source of Heat

Air appears to be handy and free to use, consequently many of the commercial heat-pump installations harvest their heat of vaporization from this source. Mechanically, the air evaporator is comparatively simple and inexpensive to provide, another point in favor of this arrangement, but these favorable characteristics can be seriously circumscribed by a host of complicating features, as we now shall see.

Limitations of the Air Evaporator. A primary problem is that of defrosting. Since the temperature of the refrigerant in the evaporator must be maintained at least 10 deg lower than the temperature of the surrounding air in order to accomplish a reasonably rapid absorption of heat, the evaporator sooner or later acquires an overcoat of frost. The effect of this frost of course, is to insulate the refrigerant from the air; evaporation slows down or stops.

To get rid of the frost, we pump hot gas from the compressor back into the evaporator, instead of sending it to the condenser; meanwhile the house, now bereft of heat, grows cooler. Naturally, if the defrosting turns out to be a stubborn business, the heat pump becomes very unpopular, indeed.

Another serious weakness of the air-evaporator design is its susceptibility to capacity fluctuation, as a direct result of changes in air temperature. As an illustration of what this means, consider a winter's day with a temperature of 36 F, and a suction temperature in the evaporator of 10 deg less than this, or 26 F.

At 26 F, Freon 12 (which is a common refrigerant in heat-pump installations) occupies approximately 1 cu ft per lb, and this pound con-

Table 13.1. Properties of Dichlorodifluoromethane (F-12)*

Sat. temp., F	Abs. press psi	Volume		Enthalpy and entropy taken from −40 F							
				Enthalpy		Entropy		25 F superheat		50 F superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0.17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86.26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0.03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04125	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0.0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.18425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	83.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106

TABLE 13.1. PROPERTIES OF DICHLORODIFLUOROMETHANE (F-12).*(*Continued*)

Sat. temp., F	Abs. press psi	Volume		Enthalpy and entropy taken from −40 F							
				Enthalpy		Entropy		25 F superheat		50 F superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	En-thalpy	En-tropy	En-thalpy	En-tropy
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.6490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	92.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204.8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

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tains approximately 81 Btu. Since our compressor runs at a constant speed, it handles a fixed number of cubic feet of vapor per hour, and consequently the heat harvested from the evaporator is measured by the Btu in each of these cubic feet.

Now suppose that the temperature drops to 0 F. We must produce a suction temperature in the evaporator of −10 F, and at this temperature 1 lb of Freon 12 is so spread out that it occupies 2.003 cu ft. At the same time each pound bears only 77 Btu back to the compressor, which, when divided between the 2 cu ft of volume, gives each of them 38.5 Btu.

Thus we see that our compressor reaps 81 Btu per cu ft of gas handled at 36 F, but can count on a mere 38.5 Btu when the thermometer reading stands at zero. In short, cooling weather decreases the capacity of such a system, just when we most need the heat. (See Table 13.1 for properties of F-12.)

The Inside Evaporator. Since we do not always find a roof or garden convenient, or even desirable, as a place to station an air evaporator, builders of the commercial cabinet machines generally encase the evaporator in the same cabinet with the other machinery, and bring in the outdoor air by means of ducts and a fan.

This arrangement poses another problem, the potential danger of overloading the fan. This danger demands our attention as the outside temperature falls; for as it grows colder, the density of air increases, and a cubic foot of it becomes heavier and heavier for the fan to handle. Thus a fan designed to deliver, say, 2,000 cfm at 20 F may discover the weight of these 2,000 cu ft to be quite unmanageable at 0 F. In the circumstances, we may burn out the fan motor, or, if the fan is of the nonoverloading type, the evaporator is starved for air. As in the case of the frosted coils, the house grows colder and colder, and both the designer and the heat pump are likely to earn most unfavorable comments.

Obviously, in selecting an air-evaporator design, we must exercise the utmost caution in sizing the installation. Air temperatures fluctuate widely in many sections of the country over any selected 24-hr period; consequently we must choose between oversizing the plant for some of the time, and overloading it at others. Perhaps the best solution is to choose a plant to meet the normal low temperature of 20 F or 10 F, whatever it happens to be, and then provide some form of supplementary heating for the subnormal days. We shall examine some of these hybrid arrangements which are operating satisfactorily later in this chapter.

The Balance of Pros and Cons for the Air Evaporator. Here are the advantages of the air evaporator:

1. Air is always available.
2. The system design is simple.
3. The initial investment is less than for either a water or earth coil arrangement.
4. Air imposes no cost save that incurred by its handling.

Limitations may be tabulated this way:

1. Defrosting can be inconvenient and annoying.
2. Sizing is a critical problem as the result of the wide temperature ranges commonly encountered. Designers and contractors can lose their reputations by a bad guess either way.
3. Severe weather may require the provision of some form of supplementary heating.
4. The air-moving fan is vulnerable to either overloading or inadequacy under certain conditions.
5. For most satisfactory results, a full air-evaporator system should not be installed in areas where the winter thermometer falls below 20 F.

Water for Heat

Since water ceases to be water at 32 F, we can be sure with a water design of having a heat source that is never below this critical temperature. This means that we can operate on a suction temperature of 22 F, or higher, a situation that simplifies our sizing problem immensely.

Actually, if water is available, either from wells or city mains, its temperature is rarely below 40 F, and throughout the winter in most localities it is even substantially warmer than this. For example, in some sections of Minnesota, the winter temperature falls all the way to −50 F, and week-long periods with the thermometer continuously below zero are common. Despite these severe conditions, heat-pump installations using water from wells report minimum water temperatures of 38 F. In the less exacting climate of West Virginia and New England, heat pumps are operating on well water ranging from 50 to 60 F.

Compare these temperatures with those encountered by an air evaporator, and the advantage of water as a source of heat becomes obvious.

Water Requirements and Suction Temperatures. Our first problem in designing a water source system is to determine the quantity of water needed. To arrive at the answer, we must evaluate various suction temperatures and also calculate the contribution of the compressor in Btu per pound of refrigerant gas.

Suppose we examine the effects of suction temperatures, assuming that our heat pump nominally rated at 5 tons has an actual capacity of 70,000 Btu. Additionally, we shall assume that the temperature of the water is 50 F.

Our first try will be with a suction temperature of 40 F, this being the smallest of the practical differentials between the heat source and the refrigerant. Now in every heat exchanger (which of course is what the evaporator really is), the largest differential of temperatures between the two fluids occurs at the entrance, in this case 10 deg. Theoretically, this differential disappears as the two fluids complete their journeys to their respective exits, and the temperature of the water should be 40 F. However, the theory in this case is not demonstrated to the bitter end, because the rate of heat transfer becomes smaller and smaller as the differential decreases, and to achieve a water temperature equal to the suction temperature, we should need a heat exchanger extending halfway to doomsday. Consequently, practical considerations result in our having an evaporator from which the water will emerge at a temperature somewhat above 40 F, probably about 44 F.

With these conditions, every pound of water passing through the evaporator will yield to the refrigerant 50 F − 44 F, or 6 Btu. Since a gallon of water weighs approximately 8⅓ lb, our heat pump can expect to harvest 50 Btu from each gallon supplied.

Now we shall see what happens if we operate on a suction temperature of 35 F. Assuming as before that we will approach this suction temperature up to about 4 deg, we will cool the water supply from 50 to 39 F, thereby reaping 11 Btu per lb. Again multiplying by 8⅓, the water will deliver 91 Btu from every gallon supplied.

These figures indicate that for every degree we extend the cooling range we can expect to recover 8⅓ Btu from each gallon of water supplied. However, we dare not drop our suction temperature lightly, as the volume of the refrigerant gas which the compressor must handle expands as the temperature falls. In effect, this means we will evaporate fewer pounds of refrigerant at a 35 F suction than we will at 40 F, since the volumetric capacity of the compressor is practically constant. In addition, the lower the temperature of the refrigerant, the lower becomes its total content of heat, which means that our saving on water may result in additional costs elsewhere.

This situation is illustrated (figures have been taken from Table 13.1) in Table 13.2.

TABLE 13.2

Suction temp, F	Volume, cu ft/lb	Latent heat of evap., Btu/lb	Total heat in gas, Btu/lb
35	0.862	66.28	82.16
40	0.792	65.71	82.71

Applying the Figures. In order to solve this problem of water temperatures and quantities, we must now decide on the temperature of the air we wish to have emerge from the condenser, this being the maximum temperature of the heating system. For this demonstration, we will assume that 118 F is satisfactory.

As the condenser is a heat exchanger, the usual laws of nature governing such apparatus must be obeyed; consequently to obtain 118 F air, we should provide a refrigerant gas temperature of at least 128 F.

Now we refer again to Table 13.1, and observe that each pound of the gas at 128 F bears 90.6 Btu. We also observe that at the condensing temperature of 118 F the refrigerant in liquid form contains 35.6 Btu.

With these figures in hand, a study of the water requirements can be accomplished.

1. With a suction temperature of 40 F, the evaporator supplies 82.71 Btu − 35.6 Btu, or 47.11 Btu per lb. These 47.11 Btu must be abstracted from the water.

2. Since the gas from the evaporator bears 82.71 Btu per lb, and later carries a total of 90.6 Btu into the condenser, the difference of 7.89 Btu represents the contribution of the compressor.

3. At 40 F suction, the evaporator provides 47.11/55 or 85.65 per cent of the total heat reaching the condenser, the difference, 7.89/55 or 14.35 per cent, being made up by the compressor.

Our first act in converting these figures into gallons of water per minute is to multiply the 70,000 Btu per hr capacity of the heat pump assumed for this demonstration by 85.65 per cent. The result, 59,995 Btu, represents the percentage of the total load which must be furnished by the evaporator. Since each gallon of water will deliver 50 Btu at a 40 F suction, 59,995/50 indicates that we shall need 1199.9 gal of water per hour, 19.9 gpm.

Now we examine the situation with a suction of 35 F. The evaporator will provide 82.16 Btu − 35.6, or 46.56 Btu, for each pound of the refrigerant. The balance of 90.6 − 82.16, or 8.44 Btu per lb, must be made up by the compressor.

At this point we consider the effect of refrigerant volumes. Since each pound of refrigerant must deliver 55 Btu to the condenser, and possesses a volume of 0.792 cu ft per lb, the heat content of each cubic foot handled by the compressor will be 55/0.792, or 69.7 Btu, when the suction temperature is 40 F. By dividing 70,000 by 69.7, we discover that the compressor is pumping 1,004.3 cu ft per hr. These 1,004.3 cu ft represent the capacity of the compressor: no matter what we do to the suction temperature, this iron heart of the system will continue to pump 1004.3 cu ft of refrigerant gas per hour.

Now at 35 F, the volume of the gas increases to 0.862 cu ft per lb; we therefore calculate the heat content of a cubic foot under these conditions by solving the division 55/0.862. The answer is 63.85.

We have only a total of 1,004.3 cu ft to work with, and at a suction of 35 F the heat capacity of the compressor becomes 1004.3 × 63.85, or 64,125 Btu. This reduction in compressor capacity may be more than a particular job can stand, in which case we must relinquish all thoughts of a 35 F suction.

In our present study, we will assume that the reduction is acceptable and will calculate the water requirements as before, but based on the smaller total load. The evaporator's share of the 64,125 Btu will be (82.16 − 35.6)/55 = 84.65 per cent, and the compressor must make up (90.6 − 82.16)/55 = 15.35 per cent.

The gallons of water per minute at a suction temperature of 35 F will be therefore

$$\frac{64{,}125 \times 84.65\%}{91 \times 60} = 9.94 \text{ gpm} \qquad \text{approximately}$$

Pumping Costs versus Compressing Costs. Since we know that we can approximately halve our water requirements by dropping the suction temperature to 35 F, the question becomes one of pumping economies versus compressing costs. Certainly we can save on the water itself if we must buy it; in that case the comparison, provided of course that the water is received at an adequate pressure, is simply the value of the water saved, evaluated against the extra operating cost of the refrigerant compressor. Occasionally we find it necessary to buy the water, perhaps from a neighboring well, and then pump it too. This arrangement tends to assist the compressor in getting more work, especially if the pipe line is a long one and the pump must overcome a considerable head.

For the two suction temperatures already discussed, let us compare the horsepower involved for both the pumps and the compressors.

With a 40 F suction we must pump 19.90 gpm of water. If a pumping head of 20 ft (the head for an actual job should be calculated) with a pump efficiency of 60 per cent and a motor efficiency of 85 per cent is assumed, the horsepower formula is

$$\frac{19.9 \times 8\frac{1}{3} \times 20}{33{,}000 \times 0.60 \times 0.85} = 0.2 \text{ hp} \qquad \text{approximately}$$

At 35 F suction, we pump only 9.94 gpm, and substituting this value in the formula, the answer is 0.1 hp, approximately.

The compressor will provide 7.89 Btu to each pound of refrigerant gas that it handles under suction conditions of 40 F. Since the total hourly weight will be 70,000/55, or 1,272.72 lb, the compressor's total contribution is 1,272.72 × 7.89, or 10,042 Btu per hr. However, we must pay also for the less than perfect efficiency of the compressor motor; thus, estimating this value at 85 per cent, the electric bill will be based on the equivalent of 10,042/0.85, or 11,809 Btu.

With suction conditions of 35 F, the compressor is called upon for 8.44 Btu, by each pound of gas that passes through it. Now the total hourly weight is 64,125/55, or 1,166 lb, which, when multiplied by 8.44 Btu per lb and corrected for motor efficiency, imposes a charge against the compressor of 11,578 Btu per hr.

In order to express these various charges in dollars and cents, we must evaluate each job on the basis of local rates for electricity. For the example, however, we will assume a rate of 5 cents per kilowatthour.

At 40 F suction, the equivalent wattage of 0.2 hp-hr is 509, and the compressor Btu equivalent is 3,460 watts. The total hourly cost then is 3,969 × 5 cents/1,000 = 19.8 cents per hour, approximately.

For 35 F suction conditions, the total cost is 18.2 cents per hour.

In this case, a 35 F suction temperature would be thoroughly justified, provided the reduction in over-all heating capacity can be accepted. Of course if supplementary heating must be added to make up for this reduction in capacity or if the compressor speed is increased, which is sometimes done in critical cases, the cost of whatever expedient is chosen must be assessed again the lower suction temperature. There is a possibility of these added charges reversing the economic balance as we have worked it out.

Quick Estimates of Water Requirements. Our exploration of water needs at two different suction temperatures indicates that there is no really quick way to arrive at an answer on which to accept any responsibility. Throughout the country, reports are obtainable on water consumptions ranging from 1.1 to more than 4 gpm per ton. In our own example, assuming that the heat pump is a nominal 5-ton machine, with a 40 F suction, we will use nearly 4 gpm per ton; with 35 F suction, less than 2 gpm per ton.

The only safe practice is to calculate water requirements on the basis of the conditions which govern a proposed installation.

Disposal of the Water. No matter where we obtain the water for a heat pump, we must find a place of disposal, once we have finished with it.

The problem is not always a simple one. The nearest sewer is perhaps our first thought, but city sewers rarely if ever have been sized to accommodate heat-pump waste, and conceivably a neighborhood of heat-pump users could cause local sewers to become inadequate almost over night. Before we can use the sewer, we must obtain municipal approval, and if our installation is a large one, using perhaps 300 or 400 gpm, this approval may not be granted, particularly if the local sewers are old and already overloaded.

A waste well is sometimes dug, but driving the well is expensive, and every plumber who ever has built a septic tank knows that dispersal of water underground can be quite difficult if the ground is low and the weather wet.

In some fortunate areas the water is simply spilled outdoors, or piped to a lake or creek. Even these practices may have their measure of trouble from the formation of ice.

The problem is a serious one, and on occasion it must be counted among the disadvantages of a water heat source.

The Water Source Balance. Here are the characteristics of water that make it attractive as a heat source for the heat pump:

1. Water provides a gratifyingly high temperature in most parts of the country throughout the year.
2. Whatever its temperature happens to be in any given locality, there will be only mild fluctuations as the result of weather changes. This uniformity helps us immensely to size our heat pump, for we can estimate fairly accurately how much heat we will have available at any given time.
3. Overloading of the compressor as a result of weather changes is a very remote possibility.
4. We can use water in severe climates.

The unfavorable features are the following:

1. Water introduces problems of pipe corrosion.
2. Driving wells for a source of supply adds substantially to the initial cost of a heat-pump installation.
3. Buying water becomes a sizable operating expense, and some cities may refuse to sell it for this purpose, if the quantity required is large.
4. Disposal of the waste water can be an irritating problem, and is a potential source of expense.

Ground Coil Plan

The Over-all Picture. Since the earth, like the air, is everywhere (although sometimes covered by oceans), it appears to be an attractive source of heat. Below the frost line, temperatures are quite uniform, year in and year out, and certainly we have no problem of waste disposal as in the case with water.

Nevertheless, a ground coil is far from ideal. In the first place, the cost of installation is considerable and, if the country is very rocky, actually may become impossible. In the second place, we do not know as yet the proper size and shape of the coil we should choose for greatest efficiency. Many studies are in progress and have been in progress for several years, but the only uniformity seen in the results is the lack of uniformity.

Heat movement through the earth is subject to several influences, not the least of which is moisture. A wet earth provides definite advantages of heat transfer, but if we design a coil for wet ground and the winter is as dry as a desert in July, results are likely to be disappointing. From a different viewpoint, the moisture can become an abomination simply by freezing. An ice barrier around a buried coil is not any different in effect than an overcoat of frost on an air evaporator—and far more difficult to melt. Ice is always possible if we carelessly permit our suction temperatures to drop below 32 F.

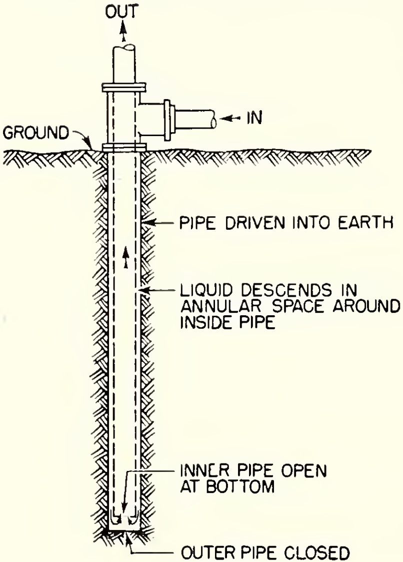
OUT
IN
GROUND
PIPE DRIVEN INTO EARTH
LIQUID DESCENDS IN ANNULAR SPACE AROUND INSIDE PIPE
INNER PIPE OPEN AT BOTTOM
OUTER PIPE CLOSED AT BOTTOM

Fig. 13.3. Ground coils, as installed at South Bend Ind.

Perhaps the best way to safeguard an installation against ice is to feed an antifreeze solution into the ground in the area of the coils. This solution will then be picked up by the subsoil moisture and diffused.

To illustrate the currently varied thinking concerning the shape and aspects of a ground coil, we have Figs. 13.2 and 13.3.

Figure 13.2 presents in principle a design that has been buried successfully in the soil at St. Louis, Mo. Eight of these springlike coils are serving a 5-ton heat pump, and approximately 110 lin ft of piping are used in each of the eight, making a total of 880 ft in all. From top to bottom, each vertical axis is 16 ft long.

In Fig. 13.3, we have a design idea that, with equal success, has been employed in South Bend, Ind. Nine of these 2-in. pipes descend 20 or 30 ft into the earth. Inside each 2-in. pipe is a 1-in. pipe, centered, and open at the bottom. In operation, a nonfreezing solution is pumped downward in the larger pipe at the bottom of which it reverses its flow, and eventually, earth-warmed, departs through the 1-in. pipe for an evaporator located more conveniently indoors.

To drive these straight pipes appears a somewhat easier task than making a coil insertion of the St. Louis style, but here again collision with some subterranean and immovable rock presents us with a more or less baffling situation.

Grid coils, laid horizontally of continuous tubing or fabricated steel pipe, similar in appearance to a radiant heating installation, are preferred by some designers. If we bury enough grid, the results may be expected to satisfy, but the cost of installation tends to be high, since we must excavate the entire area to the depth selected for the location of the piping. Then we must backfill and grade, and in some neighborhoods even landscaping may become essential.

In the present status of the art, coil design tends to reflect the economic situation; the cheapest will be favored, and may, quite logically, be the best. In fact one commercial heat-pump builder, in view of the confused regard currently being accorded ground coils, has decided that all of his models, henceforth, will be designed to harvest their heat supplies from water.

One thing is certain: Although we shall discuss the sizing of ground coils, the actual building of a coil should not be undertaken without the approval of the manufacturer who is to supply the heat pump. A careful analysis of the soil conditions before design begins is vital, and the heat-pump builders are equipped to make these analyses and to evaluate their findings.

Sizing the Ground Coil. The crux of ground coil sizing resides in the question of thermal conductivities for the soil in which we wish to bury the coil.

For the soil in which the Armour Research Foundation has prosecuted its experiments, the thermal conductivity is reported as being 1 Btu/ft^2/hr/in. of thickness per degree of temperature difference.

Based on this value, Armour has developed an approximation that is reasonably suited for similar conditions: The temperature difference between the evaporating refrigerant in the ground coil and the surrounding earth is equal approximately to the number of Btu per hour that may be drawn from each linear foot of coil.

In applying this approximation to practice, suppose that we have a ground temperature of 50 F (not unlikely below the frost line even in northern latitudes) and a suction temperature of 34 F. The difference of 16 deg becomes the number of Btu we can expect from each linear foot of the coil. If we have a heat pump rated at, say, 70,000 Btu per hr, of which the ground coil must furnish about 84 per cent, then we shall need according to this rule $(70{,}000 \times 0.84)/16 = 3{,}675$ lin ft.

This appears to be an extended bill for pipe, but we must remember that the approximation has been developed for a very definite set of conditions. Should our soil conductivity analyze at 3 Btu instead of 1, the calculated coil length immediately shrinks to little more than 1,200 ft.

There is also the effect to be considered, of pipe diameter. Currently 2-in. piping is being favored; however, both experiments and practice may indicate eventually that a larger pipe or a smaller pipe or perhaps a pipe with fins on it is actually a much better collector of Btu.

The Ground Coil Summary. Inviting features of the ground coil plan for heat harvesting are the following:

1. A ground coil provides uniform temperatures at the evaporator, thus simplifying the heat-pump sizing problem.
2. Ground generally is convenient.
3. There is no problem of waste to be solved.

On the discouraging side, we must consider the following items:

1. Ground coils can be quite expensive.
2. There is danger of ice barriers forming underground which effectively check the transfer of heat from earth to pipe.
3. In rocky areas, installation may be impractical or impossible.
4. There is insufficient design data available at the present time.
5. Two major builders of heat pumps for the commercial market do not provide ground coil designs and installations.

This summary illuminates the entrancing possibilities of ground coils for heat pumps, but it also makes clear that many problems of a practical nature must be solved before the scheme can hope for general acceptance.

Combination Designs

Purpose of Combination Designs. Since each of the air, water, and earth designs as such has definite limitations and equally definite advantages, designers sometimes attempt to cull the best features from each, while simultaneously avoiding the unfavorable characteristics, thereby producing a sort of hybrid that, like a mule, performs better than either of its progenitors under certain conditions.

For example, the simplicity and economy of air evaporators are highly desirable, but the uniform temperatures characteristic of water and earth heat are equally desirable. Consequently, a currently successful arrangement uses the air evaporator for mild weather, and then switches over to water or earth as the thermometer plunges downward. Under this plan, the air evaporator is never overloaded, and there is no buying of water or any expense for pumping during much of the heating season.

As an example of how this plan works out, suppose that we have a heat load of 60,000 Btu per hr with an outside temperature of 0 F. According to Fig. 13.4, which presents curves of capacity based on data published by a commercial manufacturer, we shall require a unit rated at 7½ hp to satisfy this load with a single compressor. Now we shall see what might be done with 2 units, only one of which uses the air evaporator. One

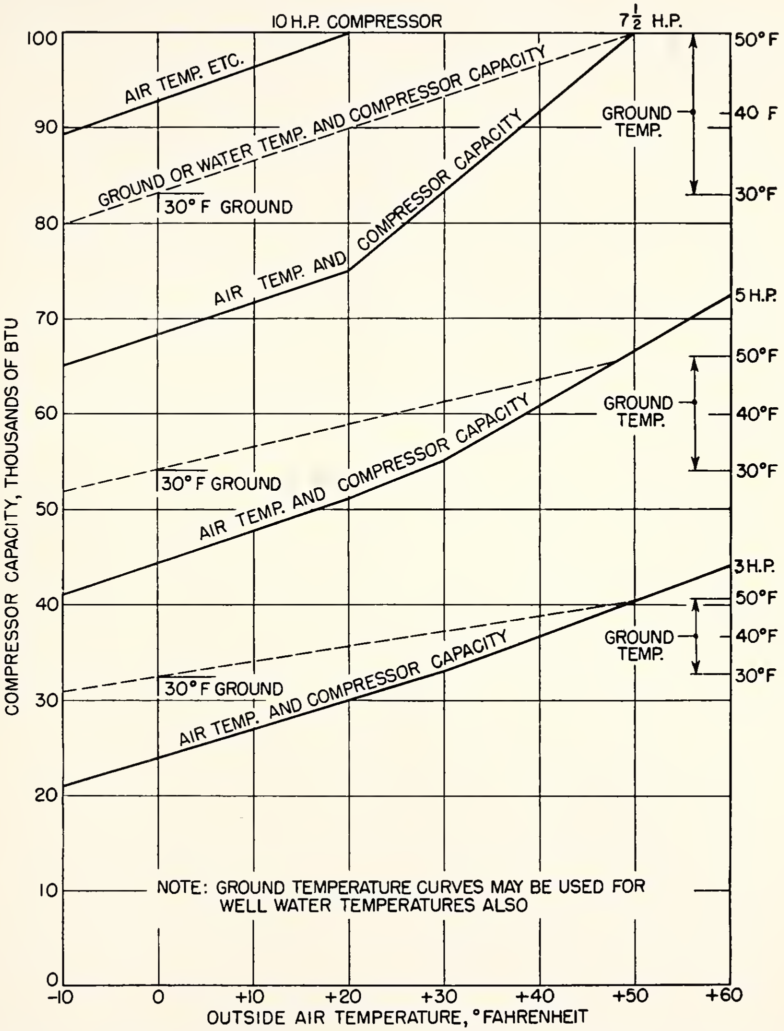

Fig. 13.4. Capacity chart with air, ground, and water as alternate sources of heat.

3 hp air evaporator will deliver 24,500 Btu per hr, according to the graph, leaving us short by 35,500 Btu. However, the difference is readily met by a 3-hp unit using a ground coil, or water as a source of heat, if these sources have a temperature 36 F, or higher.

Thus, by using two compressors with different sources of heat, we can accomplish a heating job with only 6 hp, a clear saving of 1½ hp.

The installation of ground coils in conjunction with an air evaporator is operating successfully in South Bend, Ind. The total length of ground coil, arranged in the nine 2-in. pipes we have previously discussed, is less than 300 ft.

At Abingdon, Va., a residence is employing well water at 56 F to supplement the air evaporator. The water pumped amounts to 15 gpm.

The Ohio Power Company has been carrying on heat-pump studies for several years and, among other installations, possesses one at Portsmouth, Ohio, that automatically switches over from outside air to city water, as a descending mercury crosses the 20 F level. As the mercury rises, the switch is reversed at the same point. The city water sprays on an evaporator that has been designed for that purpose.

The Heat Pump and Storage Tanks. To compensate for the shrinking capacity of an air evaporator as the weather grows colder, attempts have been made to store hot water in a tank and to keep it there until it is needed to supplement the heat pump.

The Ohio Power Company employs such a system at Steubenville, Ohio. The first weakness of the system is the size of storage tank needed, and the second is the comparatively low temperature of the water. If we operate at a condensing temperature of 130 F, for example, we shall have trouble maintaining 120 to 125 F in the tank, due to radiation and convection losses from the exterior surfaces of the tank.

Such water temperatures are suitable for panel heating, but are found in practice to be quite inadequate for service in radiators, convectors, and unit heaters. At Steubenville, therefore, the problem is solved by an electric booster on the storage-tank outlet, which steps up the temperature of the water being withdrawn to approximately 160 F.

The tank dimensions in these storage plans are potentially disturbing. For example, a 1,000-gal tank, equivalent to approximately 134 cu ft of volume, will be exhausted by a 10-gpm pump (barring replenishment) in less than 2 hr. A circulating pump of this capacity is not too large for a house of quite moderate size, and where, in such a house, can space be found for 134 cu ft of water storage?

The suggestion that we step up the condensing temperature of the refrigerant has merit, but there is little gain possible because of swiftly growing refrigerant pressures. For example, at 128 F, the pressure is 175 psig, for Freon 12. The economic ceiling of compressing and material costs is quite low; we can check it readily with a table of refrigerant characteristics.

Relative Costs of Heat-pump Installation. Despite the elimination of chimneys, coal bins, oil tanks, and ash receptacles, a heat-pump installation still costs considerably more than does an equivalent, conventional heater. For example, present market conditions indicate we can expect to invest at least $500 a horsepower for a 5-hp compressor, plus miscellaneous costs for ducts and wiring. A heat pump of this size will heat a five-room house, in a *mild climate*, with 20 F as a minimum temperature.

A comparable warm air system, oil-fired, may be expected to cost from $900 to $1400 complete, depending on the section of the country in which the installation is being made. On this basis, the heat-pump plant will require up to 2½ times the investment of a conventional system.

Operating costs likewise may involve a premium, particularly in areas of high electric rates. In Florida, electricity bills of $30 a month for 5-hp installations appear to be average. This is not a prohibitive cost at all; in fact similar heating bills received in the northern metropolitan areas would be considered reasonable, if not cheap.

From Minnesota, come figures of $103 per month during the midwinter months for a 7½-hp installation.

Operating costs can be kept down. One way is to build the house for heat-pump heating. We do this by effective insulation, preferably in the walls, roof, and under the floor. From 2 to 4 in. is recommended, the lightest being under the floor and the heaviest in the ceiling and walls. In addition, thermopane glass in all windows and complete weather stripping of the doors are highly desirable. Air temperatures from a heat-pump installation are comparatively low, from 110 to 120 F usually, whereas a conventional register emits a blast of from 140 to 160 F. Once heat is lost, therefore, with a heat-pump installation, bringing the building up to temperature is not accomplished in a matter of moments.

To summarize the economics of the heat pump, we must conclude that cheaper methods of heating are available. Nevertheless, like the automobile and airplane which are infinitely more expensive than walking, the heat pump is here to stay. We can expect its popularity to increase. Sooner or later some metropolitan development builder will provide heat pumps as standard heating systems (using their summer cooling functions as added inducements), and thereafter the heat pump will be made, so far as public acceptance is concerned. Invariably, the American public buys what it wants when it wants it, and the price, within sensible limits, is disregarded.

Summer Cooling. A heat pump sized to heat a building in winter will be found adequate in size to cool the same building in summer.

One of the problems that every designer faces when sizing an installa-

TABLE 13.3. DATA FOR COMMERCIAL PACKAGE-TYPE HEAT PUMPS*
(Using Water as Heat Source)

Compressor size, hp	Normal rating, Btu with 50 F water		Gpm required with 50 F water
	Heating cycle	Cooling cycle	
3	48,000	36,000	16
5	80,000	54,000	20
7½	110,000	81,000	25
10	160,000	120,000	25
20	300,000	240,000	67
25	375,000	370,000	84

Using Air as Heat Source

Compressor size, hp	Heating cycle, Btu, 20 F outside	Cooling cycle, Btu, 95 F outside
3	30,400	34,100
5	50,600	56,800
7½	75,900	85,200
10	101,000	114,000

* Water source units are "Marvair," as manufactured by Muncie Gear Company. Air source units are "Airtopia," by Drayer-Hanson, Inc. All data are subject to change by the manufacturer and should be verified for specific projects.

TABLE 13.4. HEAT-PUMP DATA ON FANS AND CABINETS

Compressor size, hp	Fan motor, hp	Fan capacity, cfm	Cabinet dimensions, in.			Weight, lb
			Width	Length	Height	
Water Units						
3	½	1,500	30	40	35½	1,300
5	½	2,000	30	40	35½	1,700
7½	1½	3,000	33½	46	42	1,900
10	1½	4,000	53	54	47	2,800
20	3	8,000	60	61	58	5,600
25	3	10,000	60	61	58	5,800
Air Units						
3	½	1,800*	62	48	74	1,930
5	1½	3,000	78	54	74	2,900
7½	1½	4,500	90	66	80	3,900
10	2	6,000	99	76½	83	4,800

* The figures in this column are volumes of conditioned air. Outside air is circulated over the coils in volumes approximately one-third greater, and by means of a separate fan.

tion is the frequent disparity between the seasonal requirements. This is caused by the much smaller temperature range over which the machine works in summer. We seldom cool more than 15 degrees from the outside dry bulb, but the spread on the winter heating cycle is more than 4 times 15 on numerous occasions. The machine simply is asked to provide far more Btu for heating.

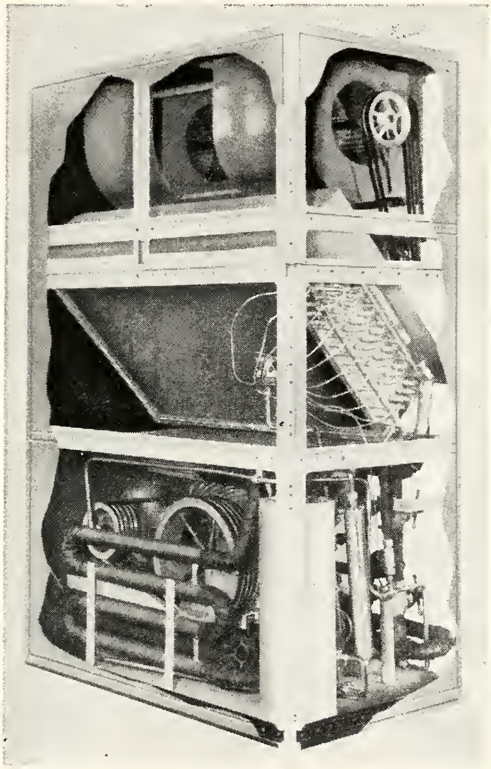

Fig. 13.5. Cutaway view of Marvair, 10-ton heat pump. (*Courtesy of Muncie Gear Works, Inc.*)

The consequence is that many installations that should be designed for heat-pump all-year service cannot stand the cost of meeting the higher heating load; therefore summer cooling is provided plus a conventional system for winter heating. This is particularly true of commercial establishments, such as restaurants and retail specialty stores.

In the residential field, the picture for the heat pump is more promising. Here summer cooling is still struggling for general acceptance, due largely to the cost involved, although many individuals do not like it. There is some evidence that home builders, who will not make the investment for summer cooling as such, will be interested in a heating system that offers summer cooling as a bonus at reasonable additional cost.

Sizes and Capacities. While heat pumps can be built in any size, the commercial builders who sell their products as packaged plants have standardized on several uniform capacities.

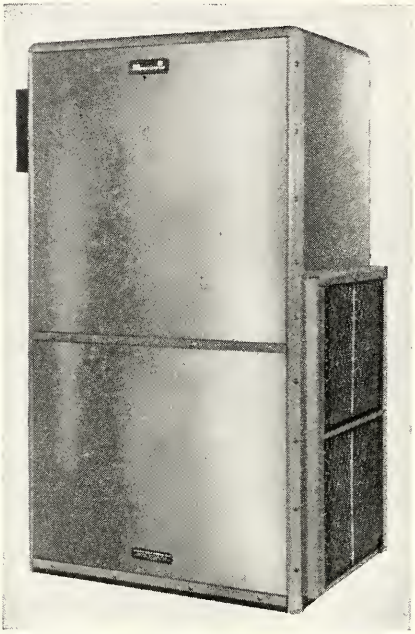

Fig. 13.6. Five-ton heat pump complete in cabinet. (*Courtesy of Muncie Gear Works, Inc.*)

Table 13.3 shows compressor ratings, Btu ratings, and cabinet dimensions. Table 13.4 provides information on the horsepower of fan motors, fan capacities, and other pertinent data.

Cabinet models are illustrated by Figs. 13.5 and 13.6.

Questions

1. Describe the principle of operation of the heat pump.
2. Name the three sources of heat currently in use.
3. What the advantages of the air evaporator?
4. What are the limitations of the air evaporator?
5. How does an inside evaporator fit into the outdoor air circuit?
6. Why is there a potential danger of the outside air fan being overloaded?

7. Calculate the heat available from each gallon of water entering the evaporator at 55 F, assuming a suction temperature of 33 F.

8. Why do we reduce the water requirements when we lower the suction temperature?

9. What is the hourly cost of operating a heat pump of 60,000 Btu capacity, using a suction temperature of 33 F and an initial water temperature of 55 F? For the cost of power, use your local rate.

10. If you were contemplating a heat-pump installation in your neighborhood using city water, how would you dispose of the waste?

11. Name three ways of handling waste.

12. Name the favorable aspects of water as a heat source.

13. What are the limitations of water?

14. What is the general status of ground coil design?

15. Compare the advantages and disadvantages of ground coils.

16. Describe a hybrid installation, using two different sources of heat.

17. What are the advantages and limitations of the storage tank?

18. Discuss public opinion and trends concerning the heat pump.

CHAPTER 14

FUELS AND FUEL HANDLING

Coal and Its Characteristics

The combustible materials of a coal are primarily carbon and hydrogen. Sulphur which is generally present will burn also, but the heat released is relatively small in importance since the sulphur is a minor ingredient.

In the bituminous coals, we find substantial compounds of carbon and hydrogen, termed hydrocarbons, identifiable in the coal gas which separates rapidly from the coal on heating. These hydrocarbons burn as gases rather than solids.

TABLE 14.1. SIZING SPECIFICATIONS FOR ANTHRACITE COAL*

Commercial size	Test mesh, in.		Round mesh			Max impurities, %	
	Through	Over	Oversize, max, %	Undersize			
				Max, %	Min, %	Slate	Bone
Broken	4⅜	3¼	....	15	7½	1½	2
Egg	3¼	2 7/16	5	15	7½	1½	2
Stove	2 7/16	1⅝	7½	12½	7½	2	3
Nut	1⅝	13/16	7½	10	5	3	4
Pea	13/16	9/16	10	15	7½	4	5
Buckwheat	9/16	5/16	10	15	7½	12% ash	
Rice	5/16	3/16	10	15	7½	13% ash	
Barley	3/16	3/32	10	20	10		
Culm	The residue after screening						

* *Note:* The sizes tabulated have been approved and adopted by the Anthracite Committee, Harrisburg, Pa.

The commercial sizes of coal, prepared for domestic fuel, vary in different localities to some extent, but Table 14.1 may be considered as representative of commercial practice. For stokers, recommended sizes are shown in Table 14.2.

Peat. Peat is the most primitive form of coal. More strictly speaking, it is vegetable matter on its way to becoming coal. We find it in bogs, sometimes looking fibrous and spongy, sometimes like a black clay.

When dug out of the bog, it contains up to 90 per cent moisture, and must be dried before it can be used for fuel Drying in the simplest form consists merely of being exposed to the sun and air in a convenient field. Mechanical drying is likely to be uneconomical, for the product, when ready to burn, offers less than half the heat value per pound than does a medium good coal. In short, considering the handling charges involved, coal is almost certain to be cheaper.

TABLE 14.2. RECOMMENDED STANDARD COAL SIZING SPECIFICATIONS*

Type of equipment	Kind of fuel	Sizing	Remarks
Traveling grate	Anthracite	No. 3 buckwheat (barley). All through a 3/16-in.-round mesh and not more than 15% through a 3/32-in.-round mesh	
	Anthracite	No. 4 buckwheat. All through a 3/32-in.-round mesh and not more than 15% through a 3/64-in.-round mesh	
	Coke breeze	All through a 5/8-in.-round mesh and not more than 50% nor less than 25% through a 1/8-in.-round mesh	Should contain 8 to 10% moisture and not less than 2% volatile matter†
Traveling grate and chain grate	Midwestern bituminous and lignite	1-in. nut and slack, not more than 50% slack through 1/4-in-round mesh	Coal should be uniformly tempered up to 15% total moisture†
Spreader	Eastern bituminous (friable)	1 1/4- or 1 1/2-in. nut and slack, not more than 50% slack through 1/4-in.-round mesh	†
	Eastern bituminous (nonfriable) and Midwestern bituminous	3/4-in. nut and slack, not more than 50% slack through 1/4-in.-round mesh	†
Underfeed stokers, single or multiple retort	Eastern bituminous (friable)	2-in. nut and slack, not more than 50% slack through 1/4-in.-round mesh	For multiple retort†
	Eastern bituminous (nonfriable) and Midwestern bituminous	1-in. to 1 1/4-in. nut and slack, not more than 50% slack through 1/4-in.-round mesh	For multiple retort†
Pulverizers	All coals	3/4-in. nut and slack	

* Courtesy of American Boiler Manufacturers Association and affiliated industries.

† Fuel to be delivered across stoker hopper without segregation.

Lignite. When peat obtains its first promotion in Nature's chemical factory, it becomes a brown to black, crumbly sort of coal called lignite.

Lignite is still rather wet stuff when mined, with up to 40 per cent moisture, but happily most of this moisture evaporates in short order when lignite is laid out to dry.

Lignite is somewhat better than peat in calorific value, but not enough to justify very extensive shipping and handling. In boiler plants convenient to lignite beds, its use is common, and when pulverized before burning, the results are reported to be entirely satisfactory.

Subbituminous Coal. Here is a low grade of bituminous coal that we must be careful not to confuse with "semibituminous" which, being very high-grade coal indeed, is the antithesis of the subbituminous variety.

Subbituminous coal tends to disintegrate rapidly when stored in the weather. It is used mostly near the point of mining.

Bituminous Coal. The nation's business is based primarily on bituminous coal. This is because bituminous is our most plentiful fuel.

Bituminous varies in its heating value over a considerable range, each field having characteristics peculiar more or less unto itself. These characteristics tend to restrict the application or service of any given coal. For example, bituminous coal that cokes well is bought for coke-oven plants and for gas manufacture.

Coking coals are generally also termed "caking" coals, which means they tend to fuse into a solid mass when heated. This group is rich in hydrocarbons, both in the form of gas and tar.

Most domestic users prefer the noncaking, or "free-burning," coals.

Nearly all the steam-making coals are bituminous, and we find them caking and free-burning, depending largely on what is available at the best price. Ash content is an important item in the coal analysis, because from the heat-liberation standpoint, its value is nil. If we buy a ton of coal that is half ash, we actually obtain half a ton, or are paying twice the quoted price per ton, depending on which way we care to look at it.

And yet ash, under some boilers, has considerable value as a reducer of maintenance. For example, coal burning on a grate under a hard-pressed industrial boiler may readily burn out the grate bars unless these iron units are protected by a blanket of ash between their surfaces and the flames.

The ash content becomes particularly critical on stoker-fired installations employing preheated air; if the ash content for this service is less than 5 per cent, trouble is almost inevitable.

Bituminous coal for industrial operations should be bought on specification. If a West Virginia coal, for example, provides the characteristics we want, then we certainly should continue to demand West Virginia coal and should refuse to accept other coals which are less satisfactory to our operations.

The importance of getting what is best for us is illustrated by the experience of a very large district heating plant that served the business areas of one of America's largest cities. This plant had been buying a suitable, high-grade coal, and then one day the plant personnel attempted to burn some low-grade nonspecification coal that had been lying around for a long time The day happened to be the coldest of the winter, with zero temperatures, and the boilers were heavily loaded, a situation that everyone agreed would make burning the poor coal easy; the furnaces seemed hot enough to burn anything.

Unfortunately the low-grade coal was a slagging coal. As it burned, it fused over the stokers, blanketed the water walls, and generally performed a strictly evil service. The steam pressure fell off, and consequently the steam-driven stokers, pumps, and fans all lost speed, swiftly making a bad situation critical. Men tried to break loose the slag with steel bars, and were overcome by heat and gas. Within an hour, half the plant personnel were in a city hospital, while the center of the city, deprived of heat, shivered and complained bitterly.

That plant never again has tried to burn nonspecification coal.

Semibituminous Coal. Semibituminous is considered to be one of the finest of all coals. It is low in moisture, ash, and sulphur, and has less volatile matter than bituminous coal. It burns freely and without excessive smoke.

Much semibituminous coal is sized and sold for domestic use.

Anthracite Coal. Anthracite is hard and smokeless. Most of it goes into domestic furnaces, although in late years it is finding more acceptance by industry as a means of eliminating smoke and air pollution.

Small stuff, practically dust, discarded by the sizing screens and called "culm," was considered for many years to be all but worthless. Every anthracite town had its culm pile looming on the horizon, sometimes almost a mountain in size.

Currently, culm is being mixed with high-volatile coals and is burned successfully as pulverized fuel.

Cannel Coal. At one time cannel coal was an important material in the making of illuminating gas. Being rich in hydrogen, which burns with a bright flame, it contributed this brightness to the product of the gas works in sufficient candle power to meet the local requirements.

Gas has lost much of its use for illumination in late years to electricity; consequently the services of cannel coal in gas making are less urgent. However, cannel coal is still a favorite on the grates of open fireplaces, where its bright flame and free-burning character are much appreciated.

Cannel coal is a form of bituminous found in a comparatively few locations particularly Kentucky, West Virginia, and Utah.

The Comparative Picture. The characteristics of the various forms of coal are tabulated in Table 14.3. These characteristics are not indicative of all bituminous coals, all lignites, all peat, or all anthracite. The only way to obtain a really dependable analysis of any coal is to obtain an analysis of the coal from the specified field. Differences occur even here, for all coal is not uniform from the same field. This fact alone points up the desirability of buying coal on specification.

TABLE 14.3. APPROXIMATE CLASSIFICATION OF COALS*

Name	Fixed carbon, %	Volatile, %	Moisture, %	Heat value, Btu/lb
Dry peat	23.0	52.0	25.0	7,100
Lignite	37.8	18.8	43.4	7,400
Subbituminous	42.4	34.2	23.4	9,700
Low-grade bituminous	47.0	41.4	11.6	12,800
High-grade bituminous	64.6	32.2	3.2	15,100
Semibituminous	83.4	11.6	5.0	15,400
Anthracite	95.6	1.2	3.2	14,400
Coke	92.5	1.6	5.9	14,000

* *Note:* These figures are given on an ash-free basis.

Coke. If the volatile matter is distilled from coal, what is left is coke. This procedure is practiced in the artificial gas plant, the coal being sealed up in a retort and heated by external fires. The gases are then bled from the retort for further purification and industrial use, while the erstwhile host ceases to be coal and leaves the retort as "gas coke."

Coke also results from the refining of crude petroleum, and is shipped from the refinery under the name of "petroleum coke."

But this is not all, for another type of coke is the result of distilling tar. This member of the family is a notably homogeneous substance, innocent of oils and waxes, and bears the name of "pitch coke."

All the cokes are high in carbon content, and some of them contain much ash too. They burn rapidly and without smoke, and in those unfortunate cases of a chimney flue being too small for the furnace, changing the fuel from coal to coke may produce satisfactory combustion.

The heating value of coke will range from 12,000 to 14,000 Btu per lb, approximately.

Coal Storage and Handling

Outdoor Storage The bituminous coals slowly oxidize when exposed to the weather; consequently after a number of years spent lying on a storage lot, a coal sample from the pile may show an appreciable reduction in heat value with a corresponding increase of ash. There is likely to occur also a breaking down in size, especially if the coal is of a low grade.

Coal-storage piles whether outdoor or indoor should be arranged to allow a periodic checking of temperatures. A pipe driven deep into the pile will serve as a passage for a thermometer; if the temperature rises above 160 F, spontaneous combustion is likely to follow.

Spontaneous combustion is the result of the oxidation process giving off heat which cannot be dispersed by a tightly packed condition of the coal. Sulphur may contribute to the danger, but its importance has not been conclusively measured. Apparently it assists combustion under some conditions, but is a noncontributor under others.

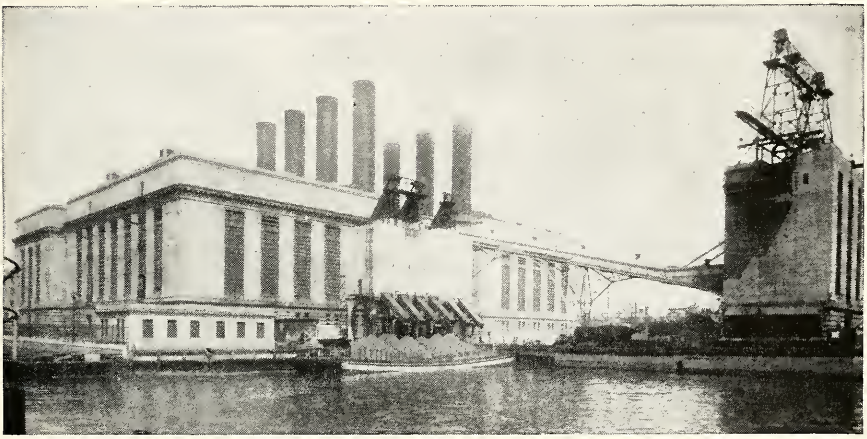

Fig. 14.1. Coal-lift tower at the Delaware Waterside Generating Station of the Philadelphia Electric Company. Shown also are the ash bin and discharge chutes. Both coal and ashes are handled by barge.

Coal has been stored under water with no adverse effect on its heating qualities according to reports. Such storage prevents spontaneous combustion, obviously, but introduces additional problems of handling and drying.

Bins and Bunkers. The compartment that holds our coal for the winter in a residence is a coal bin; increased to industrial dimensions, and perhaps mounted high in the building to feed down to the boilers by gravity, it becomes a bunker.

Domestic bins are charged usually through a cellar window, with the coal sliding down a chute inserted in the window by the coal delivery men. For industrial delivery, the coal arrives on the scene by barge, truck, or train. From the barge, it is lifted by a grab bucket, sometimes hoisted a hundred feet or more into the air, and dumped into an overhead bunker. Figure 14.1 shows a large public utility plant that operates its coal system in accordance with this plan.

Coal cars of the gondola type deliver their load of coal by opening the hopper gates in their bottoms and spilling the coal to an undertrack bin whence it is transferred to the permanent bunker by a moving belt or bucket conveyor. These transfer media are illustrated by Figs. 14.2 and 14.3.

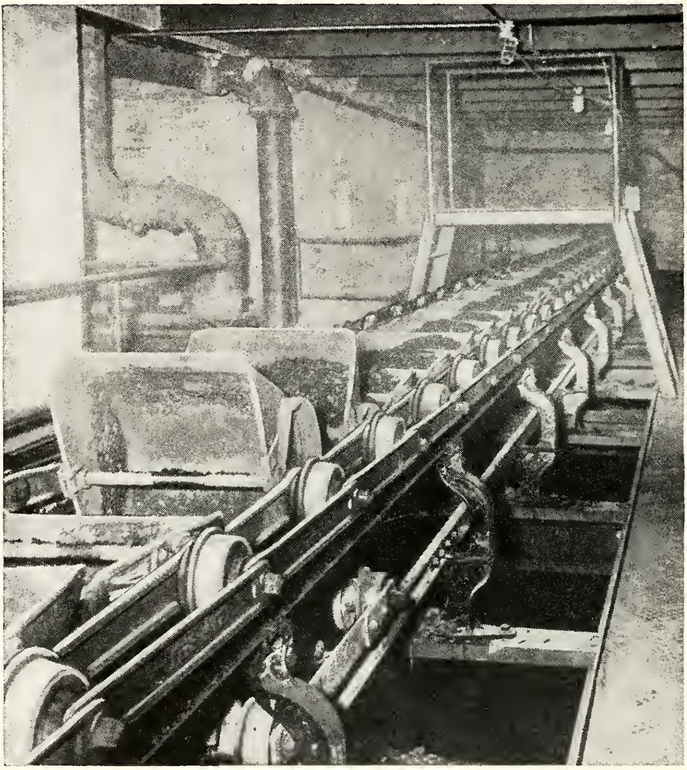

Fig. 14.2. Close-up of bucket conveyor. (*Courtesy of Link-Belt Company.*)

Another method of delivering coal from a car, but a method we are unlikely to meet in designing for heating installations since its use is justified only where enormous quantities of coal must be handled, is the car dumper. This device picks up the car and the section of track on which it stands, rotates through 180 deg which brings the car to an upside-down position, pauses long enough for all the coal to fall out of the car, and then restores the car to its normal position.

Sizing Bins and Bunkers. The domestic coal bin should be sized to store a complete season's demand for fuel. A bin of this size allows off-season buying with a consequent saving in cost.

Coal weights vary, and if space is rather tight for our bin, we should obtain the weights per cubic foot of the particular coal we plan to store. For rough estimating, some useful figures are given in Table 14.4. From this table we learn that a ton of lump anthracite occupies 2,000/55, or approximately 36.4 cu ft. If the heating plant burns 10 tons of this coal in a season, then we should provide about 364 cu ft of bin.

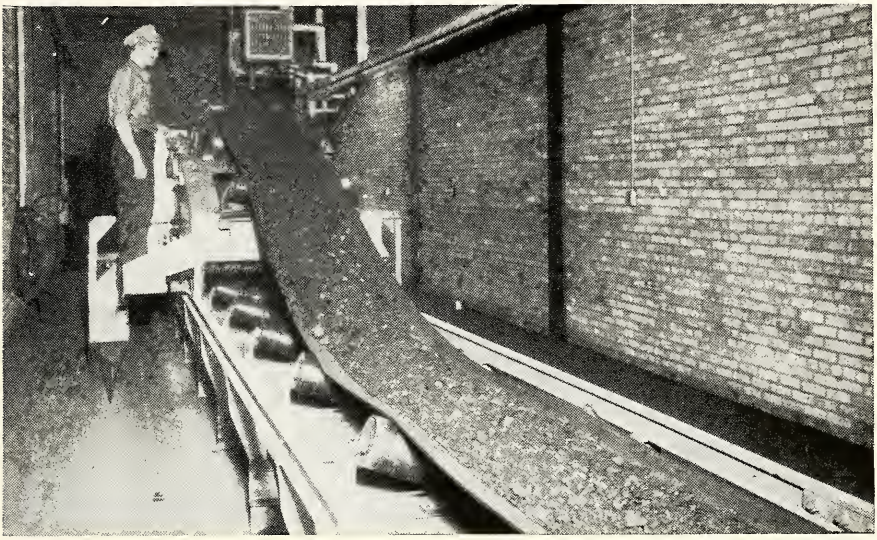

Fig. 14.3. Continuous belt conveyor. (*Courtesy of Link-Belt Company.*)

TABLE 14.4. COAL WEIGHTS PER CUBIC FOOT

Kind of Coal	*Weight, Lb/Cu Ft**
Bituminous, lump	50
Run of mine	52
Semibituminous	55
Anthracite, lump	55
Anthracite, buckwheat	51
Coke	25

* These weights are approximate, and have been compiled from a study of the weight ranges reported from various fields.

The size of an industrial bunker is decided upon after a consideration of two factors: First, what is the maximum time the plant can operate between deliveries of coal? If the boilers burn 10 tons of coal a day, for example, would 10 days, supply at a time be safe, or should a greater allowance be made for irregularities of delivery?

When the storage capacity is determined, this capacity then must be weighed against the second factor, the consideration of cost. For an overhead bunker, the great weight of the coal in storage imposes important structural and foundation problems, and the cost of solving these problems satisfactorily possibly will be greater than the budget can bear. In cases such as this, a smaller bunker with supplementary ground level storage and handling equipment must be considered.

Overhead bunkers are built of concrete or steel. Concrete increases the problems of weight, but steel is subject to corrosion and erosion. Coal in motion is an abrasive, and this characteristic poses many problems in the design of coal chutes and gates. Recently some success has been reported with hard glass liners installed in steel chutes at points of critical wear, but a glass-lined bunker could be more costly than solid concrete. A concrete liner has been used at times, and so long as the liner does not crack, it provides a satisfactory bunker.

Ash Handling

Domestic Ash. There is only one vital rule to be observed in handling the ashes from a domestic furnace: Never place ashes, either hot or otherwise, in a combustible container.

The volume of ash to be handled from residential heating is too small, generally, to justify any mechanical equipment for its handling. Occasionally architects provide an ash pit, with a cover, from which the ashes are taken at selected intervals.

Industrial Ash. Figure 14.4 illustrates a common arrangement for handling ashes from an ash alley provided beneath large industrial boilers. The ash cars in large plants are hauled away by narrow-gauge locomotives; in small plants they often are pushed about by hand or are dragged by a tractor. Figure 14.5 shows a form of ash gate.

In Fig. 14.6 we see ashes being transferred from the ash car to a storage hopper by means of a skip hoist.

To avoid the ash-car equipment, we can design our plant with sufficient headroom in the ash alley to permit a standard dump truck's taking position beneath the ash gates. Of course this extra headroom adds to the cost. We must lift the whole boiler structure, or dig a depressed roadway, and either alternative might involve a greater expenditure than the purchase of a couple of cars.

A method of ash handling that avoids both cars and extensive headroom is the hydraulic system. The ashes are simply sluiced off to their destination by water, which picks up the ash from the boiler hoppers and in the form of an ashy slurry disposes of the whole mess neatly and more or less invisibly behind the walls of special piping.

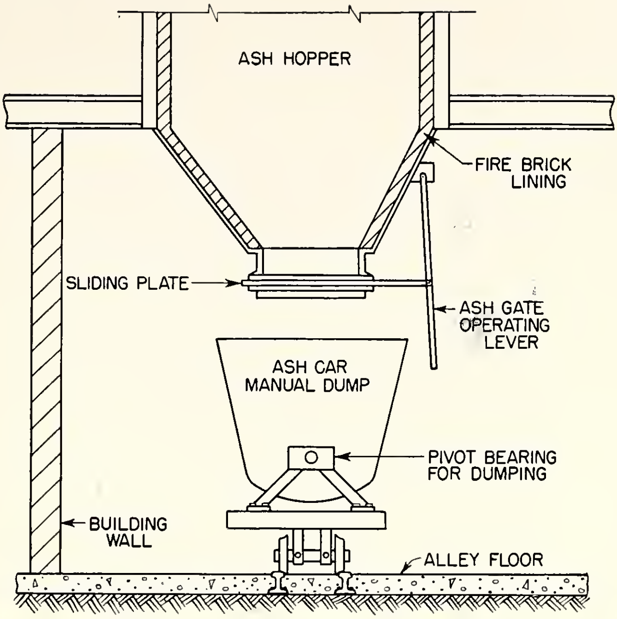

Fig. 14.4. Ash alley showing narrow gauge car under ash gate.

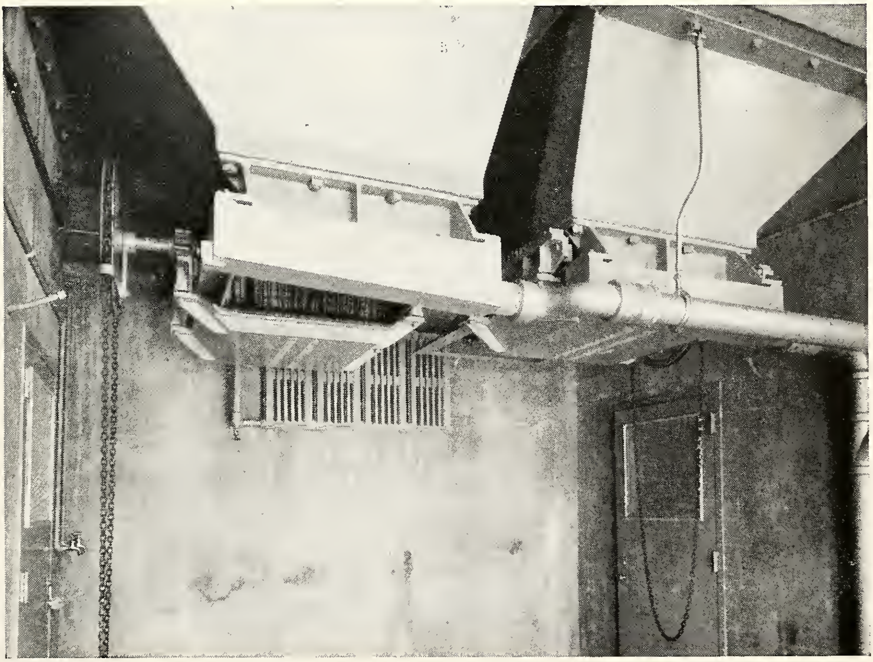

Fig. 14.5. Ash-gate installation under industrial boiler. *(Courtesy of Beaumont-Birch Company, Inc.)*

Ash, like coal, is both abrasive and corrosive, consequently maintenance on a hydraulic system can be a continuing problem. For this reason, the economics of hydraulic systems should be considered carefully before a commitment is made.

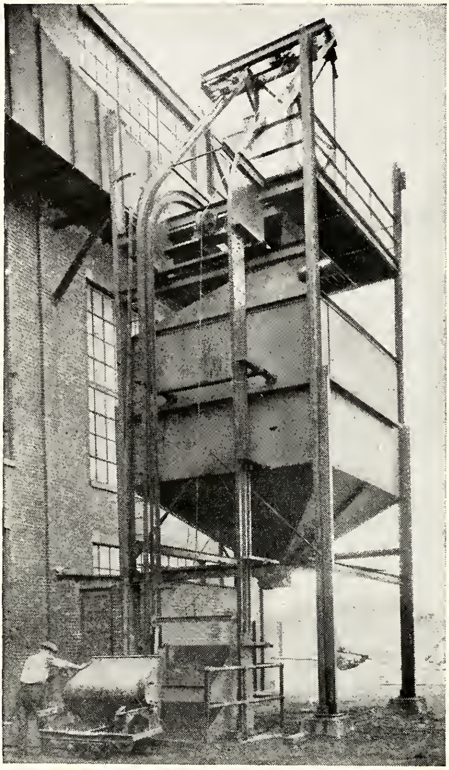

Fig. 14.6. Skip hoist installation showing ash car being dumped into skip, and skip commencing to rise to top of bin. (Courtesy of *Link-Belt Company.*)

Wood Fuels

Except for the conventional practice of burning logs in a fireplace, wood for heating becomes available generally in one of two forms. The first of these consists of roughly sized chunks, sometimes augmented with shavings, and called "hogged fuel." The second form is that of sawdust.

Burning either fuel employs the "Dutch oven" arrangement more often than otherwise, although some power-boiler installations have been reported in which the fuel is dumped on grates directly under the boiler tubes, just as though it were coal instead of wood.

The hogged fuel is created by feeding large slabs and coarse waste chunks into a machine that is equipped with big rotating knives, the duty of these knives being to hack the wood into pieces averaging ½ to 1 in. in diameter. The machine is called a "hog," thereby bequeathing its name to its offspring. Hogged fuel is rated at 200 cu ft for the equivalent of a cord of wood.

A cord is a pile 4 by 4 by 8 ft, a gross volume of 128 cu ft, of which about 70 per cent is wood with voids making up the remainder. The greater volume of hogged fuel is due to its greater number of voids.

The weight and heating value of various woods are given in Table 14.5. These figures are issued by the U.S. Department of Agriculture.

TABLE 14.5. WEIGHT AND HEATING VALUE OF VARIOUS WOODS WHEN GREEN

Wood	Weight, lb/cord	Btu/lb*
Ash, white	4,300	4600
Beech	5,000	3900
Birch, yellow	5,100	3800
Chestnut	4,900	2600
Cottonwood	4,200	3000
Elm	4,400	3600
Hickory	5,700	4100
Maple, red	4,700	3700
Oak, red	5,800	3400
Oak, white	5,600	4000
Pine, white	3,300	4200
Walnut	5,100	4100
Willow	4,600	2400

* The heating value given is an approximation, useful for preliminary studies. For exact computations, both weights and heating values must be determined under the conditions of proposed use.

Domestic Sawdust Burners. In the Pacific Northwest, where lumbering is a major industry, the Engineering Experiment Station of the Oregon State College has done considerable work on the testing and rating of sawdust burners.

Figure 14.7 illustrates the application of a sawdust burner to a residential furnace. Any furnace can be used, provided the coal grates, if any, are removed.

Sawdust burners may be obtained commercially in sizes ranging between an output of 150,000 Btu per hr and 1,000,000 Btu per hr. Grate areas vary between 1.5 sq ft and 8 sq ft.

The efficiency of a sawdust burner generally is low, from 30 to 40 per cent being a fair average. Stack draft is important for satisfactory operation, a minimum of 0.1 in. W.G. at the furnace outlet being recommended by the Engineering Experiment Station. For an estimate of burner capacity, the same authority suggests the following formula:

$$A = \frac{Q}{RHE}$$

where A = area of the grate, sq ft
Q = desired output of the burner, Btu per hr
R = burning rate, pounds of sawdust per hr per sq ft of grate
H = heating value of wood, as fired, per lb
E = efficiency, per cent

For all general purposes, the work of the Engineering Experiment Station, as reported in *Bulletin* 15, July, 1941, indicates that the formula may be simplified to

$$A = \frac{Q}{150{,}000}$$

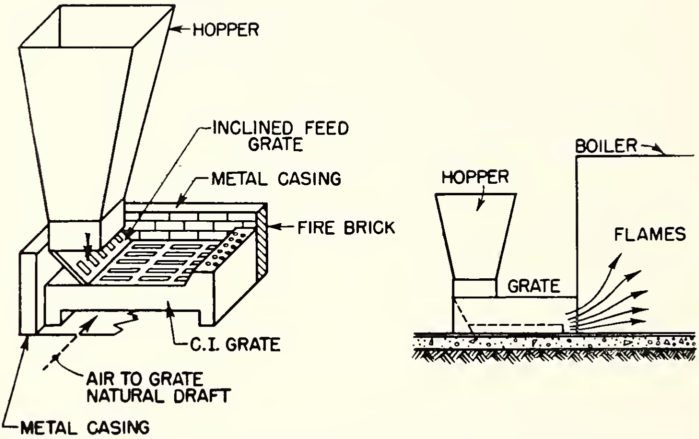

Fig. 14.7. Sawdust burner as used in northwestern United States.

Calculations for determining the products of combustion from the burning of wood are illustrated in Chap. 15.

Liquid Fuels

Commercial standard specifications for fuel oils, as recorded in the American Society for Testing Materials Designation D396-34T, include the values given in Table 14.6.

We observe that fuel oils are classified according to their characteristics in six major groups. Each of these oils functions best with some particular type of burner, and may prove completely unsatisfactory or unusable if served to a different burner.

For domestic use, grades 1, 2, and 3 are generally recommended. Grade 1, lightest of the domestic group, is suitable for the pot type of vaporizing burners, and for the pressure atomizing group.

Rotary burners perform nicely on either Grade 1 or Grade 2, although some specially designed rotary burners for industrial or commercial service are intended for Grade 3 oil without preheating. Gun-type burners also will function with the heavier Grade 3 oil.

TABLE 14.6. FUEL OIL SPECIFICATIONS

No. of grade	Min flash point, F	Max water and sediment, %	Max pour point, F	Residue, %	Max viscosity, sec
1	100*	0.05	15	0.02	
2	110*	0.05	15	0.05	
3	110*	0.10	15	0.75	70
4	150	1.00	..	....	500
5	150	1.00	..	....	100†
6	150	2.00	..	....	300†

* Or local legal limit.

† Indicates viscosity measured on Saybolt Furol instrument.

Grades 4 and 5 are sold for industrial or commercial use, and may or may not require preheating, depending on the ambient conditions of storage and pumping.

Grade 6 oil, resembling the old bunker C classification in characteristics, must be preheated prior to burning.

Let us examine some of the terms used in connection with fuel oil, and consider their meaning.

Gravity. The gravity of oil generally means the American Petroleum Institute (API) scale, in distinction from the specific gravity which concerns the relationship of the oil to water. The American Petroleum Institute has developed a standard formula for determing the relationship between specific gravity and the API scale. This will be discussed later under Baumé.

The quality of a fuel oil should not be judged on the basis of gravity alone, since modern refinery practice is producing oils in which the gravity is but partly responsible for the observed characteristics.

Flash and Fire Points. By heating an oil we cause it to emit a vapor. If the heating temperature is high enough, these vapors may separate so fast from the oil as to concentrate above the surface and ignite momentarily if a flame is applied. The temperature at which this ignition of vapors becomes possible, for any given oil, is called its flash point.

Vapor concentrations at the flash point can be ignited momentarily, but do not support combustion beyond that single flash. However, if the oil temperature continues to rise, a temperature is reached eventually at which, after the instantaneous flash occurs, the production of vapors is sufficiently rapid to establish continuing combustion. This temperature we call the fire point.

Viscosity. Viscosity is another name for fluidity—the readiness of the oil to flow. Viscosity is expressed in time, being the number of seconds required by any given oil of a definite volume to flow through a definitely sized orifice, when the oil is heated to a specified temperature.

The viscosity for domestic oils is measured on the Saybolt Universal viscosimeter with the oil going through the orifice at 100 F. For the heavier grades of oil, the viscosimeter is known as the Saybolt Furol instrument, and the standard temperature (probably to expedite the determination) is raised to 122 F. The instruments themselves are identical, except that the Furol orifice is larger.

Other systems of viscosity measurement are the Engler and Redwood. Since these have different scales, their identification by name is essential.

The Pour Test. The pour point of any given oil is the lowest temperature at which it will flow and remain liquid. Thus, if our fuel oil is said to have a pour of 30 F, we know that we cannot pump it if the temperature falls lower.

In buying fuel oil for use in cold climates, we should inform ourselves of the pour point before placing the order. Heating may be needed.

Importance of Color. The domestic oils are produced by distillation; consequently these oils should be clear and transparent. Discoloration indicates the presence of foreign materials such as rust, water, wax, gum, and carbon.

Color in the heavier oils is of small significance, and does not directly point to either good or bad burning characteristics.

Sulphur. Sulphur in fuel oil is not considered important, unless it happens to be present in some corrosive form or compound.

The Distillation Factor. Distillation tests are applied only to domestic oils, because cracking will begin above 650 F. If the end point of distillation, or boiling, is much above 650 F, a residue of carbon or asphalt may be encountered in the furnace.

The nature of the residue is determined from the field in which the oil originated. A paraffin base is characteristic of Pennsylvania oils, an asphalt base of oils from Mexico and California, and a naphthenic base of oils from coastal Texas.

With a good domestic oil, 99 per cent of the oil should boil away before the end point of distillation is reached, and at temperatures below 650 F.

Unit of Volume. In the United States the unit of volume for purchasing oils in large quantities is the barrel. A barrel contains 42 U.S. gal and weighs, net, 336 lb at the standard temperature of 60 F. There are approximately 5.6 cu ft of volume in a barrel.

Kerosene. This fuel is defined by the American Society for Testing Materials as a refined petroleum distillate with a flash point not lower than 73 F, and suitable for combustion in a wick lamp.

Kerosene generally is somewhat comparable to a Grade 1 domestic fuel oil and can be burned in pot and vaporizing burners. Typical properties of commercial kerosene are listed in Table 14.7.

TABLE 14.7. TYPICAL PROPERTIES OF COMMERCIAL KEROSENE

Specific gravity	0.8155
API gravity	42.0000 at 60 F
Flash point	132 F
Distillation end point	539 F

Kerosene for illuminating use, as specified by the U.S. Bureau of Mines, should have a flash point of 100 F and a maximum end point of distillation at 625 F. These values express the need, for illuminating duty, of a faster ignition temperature and also indicate the lesser importance of residues by approving a high end point.

The Baumé-API Scale. The hydrometer is a glass instrument that indicates the density of a liquid by the extent to which it becomes submerged when allowed to float on the liquid.

The scale on the hydrometer is a more or less arbitrary one, and must be modified by a constant in order to determine the specific gravity of the liquid being tested. On the Baumé hydrometer scale, this constant was calculated by the following formulas:

For liquids heavier than water	specific gravity $= 145/(145 - {}^\circ \text{Baumé})$
For liquids lighter than water	specific gravity $= 140/(130 + {}^\circ \text{Baumé})$

This need to figure constants on the basis of two different gravities caused some confusion; consequently the API, the U.S. Bureau of Mines, and the National Bureau of Standards (U.S.) eventually agreed on a single formula as follows:

$$\text{Degrees API} = \frac{141.5}{\text{sp gr}} - 131.5$$

The result is what we know as the API scale, and it is the only scale recommended for use in the petroleum industry. For a density equal to water, this formula produces 10 deg on the API scale. Other densities are worked out in the same way, while conversely, if we possess a value on the API scale, the specific gravity is quickly computed.

Handling Fuel Oils

The capacities of fuel oils to emit flammable vapors, plus their abilities to spread fires by the simple procedure of flowing in the direction of least resistance, has caused the National Board of Fire Underwriters to set up definite rules for their safe handling. These rules will be discussed in their general features but not in extensive detail, since designers should obtain for themselves copies of the complete rules.

Underground Tanks. Underground tanks must be buried below the level of all connecting piping. The object of this rule is to prevent the loss of oil from the tank in the event a pipe were to break, or loss from some arrangement of piping that might cause siphoning.

The earth cover on top of a buried tank should be at least 24 in. thick, although 12 in. will be acceptable if, in addition, a 4-in. concrete slab is spread on top of the earth. If the tank is buried beneath a building, the slab must go on regardless of the thickness of the earth cover.

Tanks inside Buildings. The heating designer is interested intimately in the rules for inside tanks.

1. No tank larger than 60 gal of capacity shall be located in a building above the lowest story, cellar, or basement.
2. Tanks shall not be located nearer than 7 ft horizontally to any fire or flame.
3. The maximum individual capacity for one tank is 275 gal, and not more than two such tanks shall be installed in any building unless suitably enclosed by masonry or concrete walls acceptable in design and arrangement to the National Board of Fire Underwriters.

Outside Tanks, Installed Aboveground. The minimum allowable distance between an aboveground tank and the nearest property line, or building, is specified as follows:

Group A. If the tank is equipped permanently with an approved extinguishing system, or alternatively an approved floating roof, and is used only for the storage of refined petroleum products not subject to boiling over, then the distance to the nearest property or building shall be at least equal to the greatest dimension of the tank, diameter, length or height. However, the distance need never exceed 120 ft.

Group B. Tanks in this group are not equipped with a permanent extinguishing system, or with a floating roof, and when storing refined petroleum products not subject to boilover, the distance between the tank and the nearest property line or building shall equal at least 1½ times the greatest dimension.

Oil tanks for heating systems generally will prove to be in the group B category; consequently 1½ times the greatest dimension is the distance we are likely to figure on mostly.

The distance between tanks, if the smaller of the tanks has a capacity of 18,000 gal or less, need not be more than 3 ft. For tanks larger than this, the distance shall equal at least one-half the greatest dimension.

If we are erecting the oil tanks on a stream without tide, an attempt should be made to locate the tanks on the downstream side of burnable property.

Embankments and Dikes. If the topography of the land should cause a flooding of adjoining properties as an aftermath of a tank leak or failure, then the tank or tanks must be enclosed by an earthen dike or embankment. The embankment must be high enough to contain, if necessary, the entire volume of oil in the tank. Embankment sides should slope at least 1½ to 1 and should provide a level area on the top of at least 3 ft in width.

Piping should be carried to the tank either over or under the embankment, and obviously this earthen fence must be unbroken by roads or paths.

Small tanks, each containing not over 18,000 gal, may be grouped within a single diked enclosure. Larger tanks must be enclosed individually.

Tank Construction. Underground tanks and tanks located within buildings shall be built of steel or wrought iron of the gauges listed in Table 14.8.

TABLE 14.8. SPECIFICATIONS FOR STEEL OR WROUGHT IRON TANKS

(Underground and Indoor Tanks)	
To 285 gal	16 gauge
181– 275 gal (inside buildings only)	14 gauge
286– 560 gal	14 gauge
561– 1,100 gal	12 gauge
1,101– 4,000 gal	7 gauge
4,001–12,000 gal	¼ in.
12,001–20,000 gal	5/16 in.
20,001–30,000 gal	⅜ in.
(Small Aboveground Tanks, either Vertical or Horizontal)	
Up to 60 gal	18 gauge
61– 350 gal	16 gauge
351– 560 gal	14 gauge
561–1,100 gal	12 gauge

For designing horizontal tanks over 1,100 gallons in capacity, we should be guided by the following:

If the tank is 6 ft or less in diameter, use 3/16-in. steel or wrought iron.

If the diameter is over 6 ft but less than 11 ft 6 in., it should be fabricated of ¼-in. metal or heavier.

For *vertical tanks* having capacities above 1,100 gal., the top and bottom minimum thickness are

Bottom	$\frac{3}{16}$ in.
Top or roof	$\frac{1}{8}$ in.

Shell-plate thickness is calculated by means of the following formula:

$$t = \frac{HD}{8{,}450E}$$

where t = thickness of plate, in.
H = height of tank, ft, above the bottom of the ring under consideration
D = diameter of tank, ft
E = efficiency of vertical joint in ring under consideration

Efficiency of a riveted joint is calculated in one of several ways as set forth in the ASME code, depending on the arrangement of the rivets. The National Board of Fire Underwriters sets the tensile strength of the steel used in these calculations at 55,000 psi and the shearing strength of the rivets at 40,000 psi.

For welded joints, the following formulas are used by commercial builders of tanks, with results that satisfy API requirements for thickness:

For double-welded butt-joint construction:

$$t = 0.0001456D(H - 1)$$

For lap-joint welded construction:

$$t = 0.000165D(H - 1)$$

where t = plate thickness, in.
D = tank diameter, ft
H = depth, ft, from top angle to bottom of course under consideration

The joint efficiency for the butt-welded joints is taken at 85 per cent; for the lap-welded joints, at 75 per cent. The formulas have been developed from these efficiencies.

The National Board of Fire Underwriters will accept vertical tanks with riveted shells that are constructed in accordance with the American Petroleum Institute Standard 12-A. After riveting, the tanks should be caulked, brazed, or welded for tightness.

Pipe Connections. All connections to tanks (except outside, aboveground storage tanks) should be made through the top of the tank above the liquid level. One exception to this rule is noted: For tanks of not more than 275 gal capacity, one bottom connection for gravity feed is permitted, plus one opening for an approved key-stem gate valve to facilitate cleaning or scavenging. The scavenging line must be run outdoors and capped with an oiltight cap when not in use. In any case, reinforcing of the shell is required at the point of connection.

Vent Lines and Manholes. Practice supplies one manhole, approximately 18 in. in diameter, for each tank holding 5,000 gal or more. For both underground and aboveground tanks, the vent sizes in Table 14.9 are recommended.

TABLE 14.9. VENT SIZES FOR STORAGE TANKS

Tank Capacity, Gal	*Diameter of Vent Fitting, In.*
To 500	1¼
501– 3,000	1½
3,001– 10,000	2
10,001– 25,000	2½
25,001– 50,000	3
50,001– 100,000	3½
100,001– 150,000	4
150,001– 400,000	5
400,001–1,000,000	6

The vents on all aboveground tanks which exceed 1,000 gal capacity should be protected by approved flame arresters, the arresters being accessible for examination and cleaning. In addition, all aboveground tanks except those with a cone roof in which the roof joint is weaker than the shell joints (to allow the roof to raise if excessive pressure develops in the tank) should have some form of approved pressure relief. In the event of an adjacent fire, the tank would be saved from explosion by the presence of such a relief. The type of relief is a matter of local preference, and the inspection bureau having jurisdiction should be consulted.

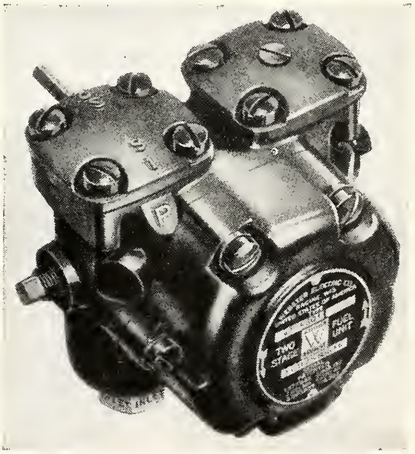

Fig. 14.8. Oil pump as employed to deliver oil from tank to burner in domestic installations. *(Courtesy of Webster Electric Company.)*

Pumps and Piping

Kinds of Pumps. Most of the oil pumps met in practice are of the positive displacement type, that is, some piston or gear actually leans against the oil and pushes. For oil burners, a gear pump with either one or two stages is commonly used.

Figure 14.8 illustrates a two-stage pump, in which the first stage picks up the oil from the storage tank, delivers it to a small reservoir built into the pump, and therein accomplishes its designed purpose. The second stage performs as a pressure pump, taking its suction from the reservoir and delivering the oil to the burner at the designed pressure for atomization.

With this type of pump, provided the overflow from the reservoir is piped back to the storage tank, it is possible to lift domestic grade 1 oil a height of 20 ft.

The horsepower for pumping oil may be determined by solving the following simple formula:

$$\text{Horsepower} = \frac{\text{gpm} \times 8.3sh}{33{,}000E}$$

where gpm = gal per min
8.3 = approximate weight of 1 gal of water at 60 F, lb
s = specific gravity of the oil at the temperature of pumping
h = total head, composed of the static or vertical head, plus the friction loss, ft
33,000 = foot-pounds, the measure of 1 hp
E = efficiency of pump, motor (in the absence of definite figures, assume 50 per cent efficiency for the pump and 85 per cent for the motor; the net efficiency then becomes $0.50 \times 0.85 = 42.5$ per cent)

Some discussion of the way we handle the specific gravity is in order.

First, the conversion formula for converting degrees API to specific gravity is based on a standard temperature of 60 F. Thus

$$\text{Specific gravity at 60 F} = \frac{141.5}{131.5 + \text{deg API}}$$

Now if we are pumping oil, not at 60 F, but say at 90 F, we must correct the specific gravity to represent conditions at 90 F. This correction involves the exercise of another formula based on plain arithmetic. Here it is

$$S_2 = S_1 \times \frac{1}{1 + 0.0005(t_2 - t_1)}$$

where S_2 = specific gravity at pumping temperature
S_1 = specific gravity at 60 F
0.0005 = coefficient of cubical expansion, average, per deg F (the oil expands on heating, this expansion being cause for its change in density)
t_2 = pumping temperature
t_1 = 60 F

To demonstrate how this all works out in practice, suppose that we assume our oil to be rated at 15 API and that we mean to pump it at 90 F. The specific gravity at 60 F is of course 141.5/(131.5 + 15), or 0.966. At 90 F it would become

$$0.966 \times \frac{1}{1 + 0.0005(90 - 60)} = 0.952$$

Centrifugal pumps are less often used for pumping oil than the positive displacement pumps, due largely to their need under some circumstances for priming. To avoid this problem, we should endeavor to locate our centrifugal pumps *below* the source of oil supply, thereby allowing the oil to flow into the pump suction by gravity.

Centrifugal pumps never overload, since the impeller simply churns the fluid if we close the pump discharge valve. For positive displacement pumps, however, we must install some sort of pressure relief valve to prevent the development of excessive pressures, this relief being mounted on the discharge side of the pump.

Piping Materials. In most of our domestic and small commercial oil burner installations, soft copper piping between the storage tank and the burner will be entirely satisfactory, since the pressure rarely approaches 100 psig. However, joints for these copper lines, particularly if the pipe is inaccessible, for example underground piping, should be made with silver solder rather than with conventional soft solder. Soft solder sometimes tends to fatigue and crack, a condition best avoided by spending a little extra installation money at the start.

Steel and wrought-iron pipe are entirely satisfactory in any size, and *must* be used when the pressure exceeds 100 psig, or the pipe diameter exceeds 1½ in. For fittings, cast-steel, malleable-iron, or welding fittings are recommended. Cast iron can be used, but appears to be more susceptible to failure than the other forms of iron. Since a disastrous fire may result from an oil-line failure, prudence in the design and construction of such lines becomes essential.

Leaks may be reduced in their degree of potential danger by burying the pipe underground as extensively as possible. However, we must protect underground pipe from corrosion, the usual method being to paint the pipe with a bitumastic corrosion-resistant material. Sprinkling the pipe trench liberally with slaked lime has provided protection in some soils. Cinder fill should be avoided, and only clean gravel employed for this purpose.

If there is some possibility of the underground piping lying full of oil but unused for any substantial length of time in cold weather, then we must install some heating means to prevent the oil from becoming so viscous that the pumps cannot obtain their suction. Running a small steam line against the side of the oil pipe for the entire length of the oil pipe is standard practice in solving this problem. The designer should provide similar steam piping for all other oil lines that may be adversely affected by low temperatures.

Preheating Oil. Figure 14.9 illustrates a conventional type of oil heater used for making heavy oils easier to handle. These heaters, which use steam or hot water for the heating agent, may be located wherever the arrangement of the system puts their services to the greatest advantage.

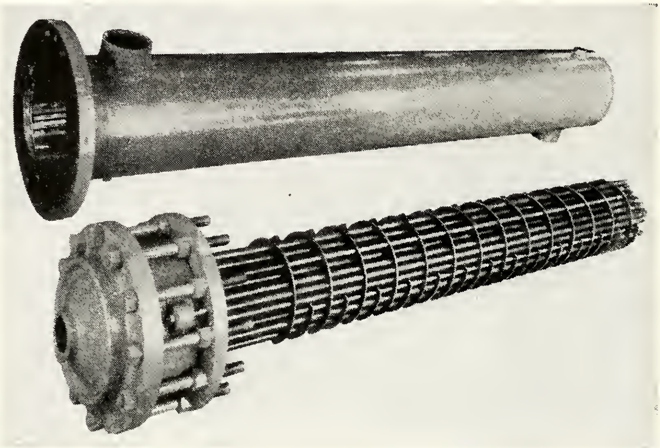

Fig. 14.9. Tubular fuel oil heater using steam. Note that oil passes around the outside of the heating tubes as guided by plate baffles. (*Courtesy of Davis Engineering Corporation.*)

In some cases, two or more heaters may be required on a single extensive run. For example, we may require one heater at the storage tank to enable us to get the oil started, and another somewhere along the pipe line to prevent the oil from cooling too much for efficient handling in transit.

Homemade heaters are common sights; but the designer will achieve more satisfactory results if he specifies the amount of heating needed to a commercial heater manufacturer, and buys a heater ready-made and guaranteed to meet the needs of the system.

Sizing Oil Piping. To size an oil piping system properly, the designer must resign himself to a rather tedious exercise in mathematics, which involves on occasion a measure of trial and error.

Oil being a viscous fluid, we must start out in any given problem by discovering the specific gravity of the oil being handled and also the absolute and kinematic viscosities.

We know how to find the specific gravity at the pumping temperature, having recently solved such a problem for 90 F oil, but the viscosity determinations introduce some new thoughts.

Kinematic viscosity is defined as the ratio of the absolute viscosity to the density of the fluid, or

$$\text{Kinematic viscosity} = \frac{\text{absolute viscosity}}{\text{specific gravity}}$$

The viscosity readings on the Saybolt viscosimeter can be converted into kinematic viscosity by solving an equation which is given by Kent as follows:

$$\text{Kinematic viscosity} = 0.00000237t - \frac{0.00194}{t}$$

$$t = \text{Saybolt readings, sec}$$

Kent also provides equations to be used similarly for converting Engler and Redwood viscosimeter readings into kinematic viscosity, and for adaption to the metric system as well.

A somewhat different formula is offered by Nelson in "Petroleum Refinery Engineering." It is

$$\text{Kinematic viscosity} = 0.219t - \frac{149.7}{t}$$

The difference in the formulas is merely in the multipliers; the t in the Nelson equation remaining unchanged as the Saybolt reading in seconds.

We may use whichever formula we wish, but the Nelson formula involves fewer numbers, and consequently will be used in working out illustrative problems.

The *kinematic viscosity* is expressed in a measure termed "centistokes" which we should know, but will seldom use by name, since most of our dealings occur with the figures and not the name.

A centistoke is $\frac{1}{100}$ of a stoke, and a stoke is the unit of kinematic viscosity. A stoke refers to a fluid having a viscosity of 1 poise and at the same time a density of 1 gram per cubic centimeter. The name honors Sir George G. Stokes, a physicist and mathematician who lived from 1819 to 1903.

Incidentally, the term "kinematics" means motion in itself, as separated from the cause of the motion. Consequently, in order to compare motion to motion, we must have some standard motion to start from, and in this case the standard is the motion of the stoke.

The *absolute viscosity* is defined as the force required to move 1 square centimeter of the fluid in the form of a plane surface, in a direction parallel to a second plane surface which is 1 centimeter distant from the first, at a relative velocity of 1 centimeter per second.

The unit of viscosity is the "poise," which is expressed as being equal to 1 dyne-centimeter per second per square centimeter. The centipoise, or $\frac{1}{100}$ of a poise, is the figure we will work with mostly in our viscosity problems. For example, the viscosity of water at 68 F is equal to 1 centipoise.

The absolute viscosity of petroleum oils varies with changes in temperature just as specific gravity varies with temperature, that is, the viscosity decreases as the temperature rises.

The importance of the absolute viscosity is that it leads us to something else, namely, a mathematical unit called the "Reynolds number." Back in 1874, Professor Reynolds discovered that various relationships of viscosity, diameter, and density, when put together properly produced this term Reynolds number, an item that is essential for calculating the friction loss of fluids flowing in a pipe.

In formula, the Reynolds number is

$$\text{Reynolds number} = \frac{Dvs}{z}$$

where D = diameter of pipe, in.
v = velocity of flow, fps
s = specific gravity of the oil at temperature of handling
z = absolute viscosity, centipoises

The Reynolds number will indicate if our flow is in the streamline, critical, or turbulent categories. The critical state falls between the other two, and is to be avoided, since at a critical velocity the friction loss is disproportionate to the volume of oil flowing. In other words, if the Reynolds number indicates we are in the critical range, good judgment suggests that we either decrease the velocity to fall back into the streamline state or else speed it up to produce turbulent conditions.

The critical range is shown on charts that are plotted on logarithmic paper, with the friction factors appearing as ordinates and the Reynolds numbers as abscissas. Since the Reynolds number is neither dimension nor measure, but purely a mathematical ratio, different designers will use different values. In Kent, for example, Reynolds numbers will appear in the thousands; in the Unwin booklet "Piping of Industrial Fuels," published by *Heating and Ventilating*, the Reynolds numbers range from 0.02 up to 100. A third investigator, F. L. Snyder, in his book "The Flow of Fluids," obtains his Reynolds numbers by a different formula;

he uses

$$\text{Reynolds number} = \frac{m}{ZD}$$

where m = flow of fluid, thousands of pounds per hour
Z = absolute viscosity, centipoises
D = inside diameter of pipe, in.

The Reynolds numbers resulting from this relationship range up to and beyond one hundred.

Figure 1.2 in Chap. 1, from Unwin, shows the critical velocity as falling between 0.1 and 1.0 of the Reynolds numbers.

The Reynolds number determines the friction factor we should use in the pressure-drop formula:

$$p = \frac{0.323fsv^2L}{D}$$

or, if we prefer to use the gpm in the formula:

$$p = \frac{0.0538fsLQ^2}{D^5}$$

where p = pressure drop, psi
f = friction factor
s = specific gravity of the oil at pumping temperature
v = velocity, fps
L = equivalent length of pipe, ft (the equivalent length equals the actual length plus the allowances for valves and fittings)
D = diameter of pipe, in.
Q = flow of oil, gpm

These formulas for viscous or streamline flow have been shortened somewhat into the Poiseuille equation of

$$p = \frac{0.00668zvL}{D^2}$$

Some designers prefer the Poiseuille formula; others take the longer way around. For practice, in order that we may observe first hand each step in the sizing of piping, we shall exercise the long formula for pressure drop, but for those who wish to investigate the design procedures more thoroughly, the illustrations should be worked out with each of the alternate formulas we have available.

Suction Piping. Since our first problem is to provide piping to the suction side of our pump, we will investigate the size of pipe required for the following conditions:

1. Pump capacity, 60 gpm.
2. Maximum allowable suction lift, 20 ft.
3. Temperature of oil, 90 F.
4. Viscosity at 60 F, 66 (SSU) seconds Saybolt Universal.

By referring to a pump catalogue, we find the suction opening on our pump to be 2½ in. Considering the height of the lift, prudence suggests that we assume a size for the suction pipe (or tail pipe as it sometimes is called) somewhat larger than 2½ in. Suppose that we start with a 3-in. pipe and see if the pump can establish suction.

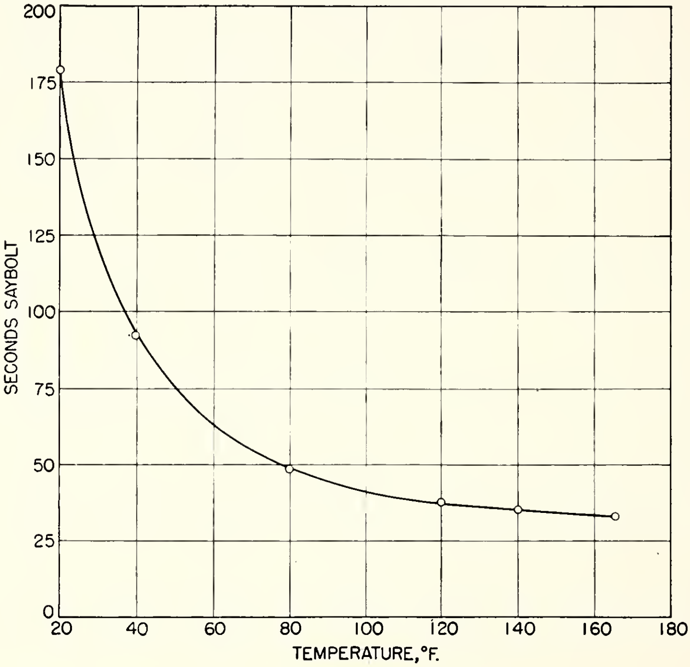

Fig. 14.10. Viscosity and temperature curve.

Our first act will be to obtain the viscosity of the oil at 90 F. Referring to the curve (Fig. 14.10), we find it to be 44 SSU. This curve was drawn from figures determined by actual test on the Saybolt viscosimeter; if we have three points, we can draw a curve.

However, oils have different characteristics, and we should obtain a curve for the particular oil we are handling. The American Society for Testing Materials has set these characteristics down in a family of curves under the title Standard Viscosity-Temperature Chart. We should obtain a copy of this chart in lieu of the actual figures, for use in any specific problem. Meanwhile, Fig. 14.10 will serve satisfactorily to illustrate the manner of application.

Step 2 requires us to determine the absolute viscosity of the oil. Employing the kinematic viscosity formula, we get

$$\begin{aligned}\text{Kinematic viscosity} &= 0.219 \times 44 - \frac{149.7}{44} \\ &= 6.24\end{aligned}$$

Changing this into absolute viscosity is accomplished by the simple multiplication of the kinematic viscosity by the specific gravity of the oil at the handling temperature. In this case, the temperature is 90 F, and the specific gravity as worked out for this temperature previously in order to demonstrate the method is 0.952. Consequently, the absolute viscosity equals 6.24 × 0.952, or 5.94 centipoises.

Step 3 brings us to a consideration of the Reynolds number. Now the velocity of the oil in the pipe exercises its influence, and we must calculate this velocity in feet per second through the 3-in. pipe. At 60 gpm, we have 1 gps, and since 3-in. pipe has a nominal inside diameter of 3.068 in. in the standard weights, the velocity becomes 2.6 fps, approximately.

By setting down all the figures we need, and have, for solving the Reynolds number formula, we obtain this result

$$\begin{aligned}\text{Reynolds number} &= \frac{3.068 \times 2.6 \times 0.952}{5.94} \\ &= 1.28\end{aligned}$$

Referring to the chart (Fig. 1.2), we see we are in the turbulent flow sector, with a friction factor of approximately 0.0075.

Step 4, which follows immediately, discloses the friction drop for any length of pipe we select. Suppose that in this case we choose a loss per foot; then we can readily evaluate the maximum suction lift our pump can handle. Thus

$$\begin{aligned}\text{Pressure drop per foot} &= \frac{0.323 \times 0.0075 \times 0.952 \times 6.76 \times 1}{3.068} \\ &= 0.051 \text{ psi per lin ft of pipe}\end{aligned}$$

To convert pressure per square inch into feet of head, we multiply the pressure by 2.42, since 2.42 ft equals 1 lb of pressure; consequently our 0.051 psi becomes 0.12 ft, approximately. The rule is $\frac{2.3}{\text{Sp. Gr.}}$; in this case $\frac{2.3}{.952} = 2.42$

At this point, we must make an allowance for the condition of the pipe surface. If it is pitted, rusty, uneven, or otherwise injects resistance to fluid flow, the allowance will be high; in some cases bad corrosion has been known to double the resistance. Some surface condition factors that practice indicates to be sound and safe are given in Table 14.10.

TABLE 14.10. SURFACE CONDITION FACTORS FOR PIPES

Smooth pipe of brass or lead	0.90
New steel or iron pipe	1.00
Wooden or concrete pipe	1.25
Vitrified pipe	1.60
Steel pipe, badly corroded	2.00
Condition unknown, but pipe new	1.30

For the case in hand, we will add the value for condition unknown, but new. The reason for doing this is that a year or two after installation, the pipe surface will have deteriorated to some extent, and if we cut our figures too fine to start with, trouble may develop with time.

Multiplying 0.12 by 1.3, our computed friction loss will be 0.156 ft. For a pipe 17 ft long, the total friction loss becomes 2.652 ft, and if we add this to the 17 ft, the total suction lift, both static and frictional, is 19.652 ft.

Apparently for the conditions assumed, we can use a 3-in. suction pipe if it is not more than 17 ft long. Should we be faced with the necessity of providing a 20-ft suction pipe, the situation would be critical, and we should be obliged to recalculate the job, using 3½- or 4-in. pipe. Situations like this illustrate what is meant by trial and error. We cannot be absolutely sure of a pipe's correct sizing until we complete the figures.

Equivalent Pipe Lengths. Bends, elbows, valves, pipe enlargements and contractions, all have the power to introduce friction into a piping arrangement. Many investigators have endeavored to measure these losses with accuracy, but, unfortunately, the conditions of flow are so ephemeral that what is true one minute is untrue the next. The best we can do is to develop some approximations. If no exact figures are at hand, the figures in Table 14.11 may be used, with some assurance that the results will be satisfactory for all practical designing in the heating and industrial field.

A paradox appears in the equivalent pipe diameters shown for the bends. Apparently, by increasing the radius of the bend, we increase the friction loss, which seems to be contradictory to current thinking. Actually, however, if we solve the fraction $\frac{\text{pipe diameters}}{\text{length of bend}}$, we shall discover that the relative loss in the short radius bend is almost double that of the longer radius bend.

Pump Discharge Piping. The same procedures that produced a suitable size for our suction pipe will also produce a suitable size for our pump discharge piping.

First, for the quantity of oil flowing, we determine the friction loss per linear foot of the assumed pipe size. Frequently, by assuming a size equal to that of the pump discharge connection, we will come up with a satisfactory friction loss on the first calculation. This is because the pump builder already has made extensive investigations as to what constitutes efficient friction factors for fluids leaving his pump, and the result is closely related to our own objectives.

Table 14.11

Item	*Pipe Diameters*
Gate valve	6
Globe valve	90
Angle valve	80
Check valve	80
Standard 90-deg elbow	24
Standard tee, side outlet	70
Standard 45-deg elbow	17
Bend (radius/diameter = 5)	23
Bend (radius/diameter = 10)	28

With average friction loss per foot known, we multiply this average by the number of *equivalent* feet in the longest run. This result becomes the total friction loss for the piping as arranged, and should be expressed in feet of head.

Next, we determine the total static head, which is accomplished by the simple procedure of measuring the difference in level between the pump and the highest point to which the oil must be pumped. This measurement is also expressed in feet.

Now we add together the static head in feet and the friction head in feet, and the result is the total discharge head in feet. This total discharge head becomes the h in the horsepower formula, by which we determine the horsepower needed to do our pumping under the conditions being considered.

As to whether the pipe size chosen is satisfactory, we must judge for ourselves by evaluating it against the horsepower for pumping, which, of course, reflects the friction loss in the piping. If we feel that the horsepower is excessive, then we must try a larger size of pipe, always remembering that larger pipe costs more money than smaller pipe, and the extra cost of the larger pipe may completely nullify the reduction in pumping cost.

Pipe-sizing Nomographs. A nomograph is a chart on which the relative values of independent variables are correlated by a straightedge.

On the pipe-sizing nomographs, all the variables of quantity, velocity, density, Reynolds numbers, viscosities, etc., may be swiftly pursued one after the other by a ruler laid on the chart. On reaching the end of the chase, we are rewarded by finding the friction loss per linear foot for any pipe size we may have assumed at the start.

Piping designers use these nomographs almost exclusively; the saving in time and the insurance against mathematical errors that may seriously affect the working out of tedious formulas make the use of these charts very desirable.

Figure 14.11 illustrates a typical nomograph for pipe sizing. Similar charts may be obtained in large sizes, for practical everyday use, from various manufacturers of pumps, pipe, and fittings. If we have much pipe sizing to do, the use of a nomograph is recommended.

Gas Fuels and Their Handling

Kinds of Gas Fuels. In the heating field, we will meet all the following gases: (1) natural gas, (2) artificial gas, and (3) liquefied petroleum gases.

Natural gas originates in many areas of the earth, and though once regarded as a resource that would soon be used up, the supply currently shows no evidence of early exhaustion. In fact, pipe lines are being extended from the natural gas fields to cities far distant, thereby increasing the use of this natural commodity.

Artificial gas, originally manufactured by the distillation of coal in a closed retort, now is largely made by the carbureted water gas process, and is a common fuel in many large cities.

Liquefied petroleum gases, sometimes called "bottled gas," are composed either of propane or butane. They are delivered as liquids under pressure to the storage tank, and a pressure controller permits them to gasify as they leave the tank.

These gases will be discussed in detail, particularly in pipe-sizing problems.

Natural Gas. The characteristics of natural gas vary as the result of the gas originating in various fields. For example, some characteristics reported in *U.S. Bureau of Mines Technical Paper* 158 are given in Table 14.12.

Natural gas normally is distributed to a community by a public utility company, sometimes in conjunction with an artificial gas system. Since the specific gravity of a carbureted water gas is about 0.64, it comes well within the range of the natural gases, and the two gases are not too different in their handling problems.

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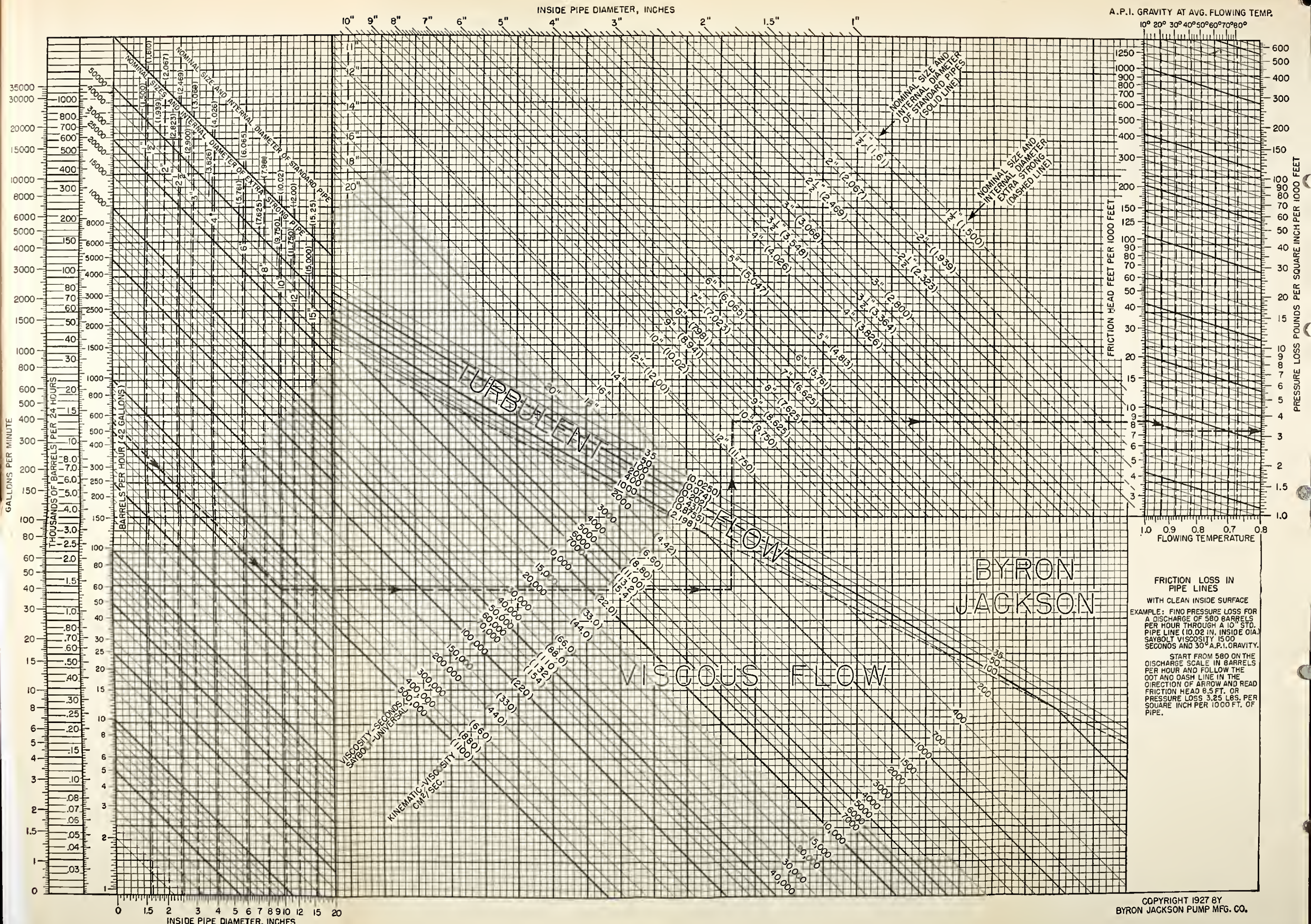


FRICTION LOSS IN PIPE LINES

WITH CLEAN INSIDE SURFACE

EXAMPLE: FIND PRESSURE LOSS FOR A DISCHARGE OF 580 BARRELS PER HOUR THROUGH A 10" STD. PIPE LINE (10.02 IN. INSIDE DIA.) SAYBOLT VISCOSITY 1500 SECONDS AND 30° A.P.I. GRAVITY.

START FROM 580 ON THE DISCHARGE SCALE IN BARRELS PER HOUR AND FOLLOW THE DOT AND DASH LINE IN THE DIRECTION OF ARROW AND READ FRICTION HEAD 6.5 FT. OR PRESSURE LOSS 3.25 LBS. PER SQUARE INCH PER 1000 FT. OF PIPE.

COPYRIGHT 1927 BY
BYRON JACKSON PUMP MFG. CO.

INSIDE PIPE DIAMETER, INCHES
10" 9" 8" 7" 6" 5" 4" 3" 2" 1.5" 1"
A.P.I. GRAVITY AT AVG. FLOWING TEMP.
10° 20° 30° 40° 50° 60° 70° 80°
GALLONS PER MINUTE
THOUSANDS OF BARRELS PER 24 HOURS
BARRELS PER HOUR (42 GALLONS)
NOMINAL SIZES AND INTERNAL DIAMETER OF STANDARD PIPE
NOMINAL SIZE AND INTERNAL DIAMETER OF EXTRA STRONG PIPE
NOMINAL SIZE AND INTERNAL DIAMETER OF STANDARD PIPES (SOLID LINE)
NOMINAL SIZE AND INTERNAL DIAMETER EXTRA STRONG (DASHED LINE)
TURBULENT FLOW
VISCOUS FLOW
VISCOSITY-SECONDS SAYBOLT-UNIVERSAL
KINEMATIC VISCOSITY CM²/SEC.
FRICTION HEAD FEET PER 1000 FEET
PRESSURE LOSS POUNDS PER SQUARE INCH PER 1000 FEET
1.0 0.9 0.8 0.7 0.8
FLOWING TEMPERATURE
BYRON JACKSON
0 1.5 2 3 4 5 6 7 8 9 10 12 15 20
INSIDE PIPE DIAMETER, INCHES

Fig. 14.11. Nomograph of flow characteristics for pipe sizing. (Courtesy of Byron Jackson Company.)

Delivery may be made either by high pressure or low pressure, this consideration being one of importance in designing the piping system.

Artificial Gas. As manufactured today, artificial gas comes from the distillation of coal in a retort, from a water gas generator in which steam is dissociated in a hot coke bed, as a by-product of coke manufacture, and from the conversion of oil.

The user seldom is conscious of the source of his gas, since it is distributed by a utility company that sends it forth into the mains with characteristics that are closely regulated by law.

TABLE 14.12. TYPICAL NATURAL GASES*

Place where gas was sampled	Methane, %	Ethane, %	CO_2, %	Specific gravity (air = 1)	Btu/cu ft†
Alabama	97.6	0.00	0.30	0.57	983
Arkansas	96.7	0.00	1.00	0.57	974
California	77.5	16.00	6.50	0.70	1062
Illinois	95.6	0.00	0.50	0.58	969
Kentucky	76.4	22.60	0.00	0.67	1167
New York	88.1	11.50	0.00	0.61	1090
Oklahoma	73.5	18.4	0.00	0.68	1064
Pennsylvania	88.0	11.2	0.00	0.61	1084
	32.3	67.00	0.00	0.89	1505
West Virginia	82.0	17.00	0.00	0.64	1125

* The difference between the total of the constituents shown in the table and 100 per cent represents nitrogen.

† Btu value at 60 F.

In gas produced by coal retorts and by the coke oven, methane is the chief combustible, and will comprise up to 48 per cent of the gas, by weight. Carbon dioxide may be expected to range from 15 to 20 per cent, and hydrogen from 8 to 10 per cent, all by weight.

In the carbureted water gas fuel, carbon monoxide may account for 45 per cent of the gas by weight, hydrogen for 4 per cent, and methane for 10 per cent. Illuminants, generally made up of olefins, with a weight of 18 per cent, are approximately double the weight they represent in the coal gases.

In calorific value, the coal gas may average around 565 Btu per cu ft; the water gas, 560 Btu per cu ft.

Oil gas is similar to retort coal gas, and is made by the breaking down of petroleum in apparatus that resembles in operation the apparatus for making carbureted water gas. In the process, oil is sprayed into a vertical cylindrical shell and there burned, with the products of combustion passing into a similar but larger shell that is more or less filled with checker brick.

The burning of the oil heats the checker brick, and when the temperature of this brick has been raised from 1800 to 2100 F, the oil flames are shut off, all air is prevented from entering the structure, and oil is sprayed into this closed and heated chamber.

The oil gasifies and continues to gasify until the bricks cool sufficiently to make the gasification too slow for economy, at which time the burners are relighted and the heating precedure is commenced anew.

In areas where petroleum is available at better prices than coal, artifical gas is almost certain to be of the oil-gas type.

Liquefied Petroleum Gases. Butane and propane are the commonly used forms of the liquefied petroleum gases for heating installations. They originate in natural gas, oil, and refinery operations. Since they are delivered to the customer as liquids and burned as gases, we have two specific gravities to consider, namely, the liquid form and the gaseous state. The characteristics of these gases are shown in Table 14.13.

TABLE 14.13. CHARACTERISTICS OF LP GASES

Item	Propane	Butane
Boils, atmospheric pressure, F	−51	15
Weight, lb per gal, liquid, 60 F	4.24	4.84
Vapor pressure at 100 F, psig	195	65
Gas, cu ft per gal	36	32
Heating value:		
Btu per cu ft	2,550	3,200
Btu per lb	21,650	21,500
Btu per gal	91,800	102,400
Specific gravity:		
Liquid	0.509	0.576
Gas	1.52	2.00

Handling Liquefied Petroleum Gases. Since the storage of liquefied petroleum gases on or near the premises where it is to be used is common practice, the National Board of Fire Underwriters has prepared regulations to be followed in handling it, intended to safeguard life and property.

First, the gas must be provided with an odorizing agent to give warning of its presence. Normally, neither propane nor butane has an odor, and concentrations of only 2.15 per cent and 1.55 per cent, respectively, by volume in an air-gas mixture, produce a combustible mixture.

In domestic and commercial systems which include a storage tank of less than 1,200 gal of water capacity, the tank must be tested and approved by the Underwriters' Laboratories, Inc., or by an equally responsible agency.

If the tank is sized for more than 1,200 gal of water capacity, then each piece of accessory equipment, including regulators, valves, vaporizer units, and relief valves, must also bear the approval of the Underwriters' Laboratories, Inc., or equivalent approval.

Containers and the first-stage regulating equipment should be located either outdoors or in buildings especially devoted to this service at distances from other buildings in accordance with the figures given in Table 14.14.

Table 14.14. Clearances for LP Gas Tanks

Capacity of tank, gal of water	Min distance, ft		Min distance between aboveground containers
	Underground	Aboveground	
Up to 125	10	None	None
126– 500	10	10	3
501–1,200	25	25	3
Over 1,200	50	50	5

Pipe Materials and Pressures. Piping or tubing may be of steel, wrought iron, brass, or copper, but it must be suitable for a working pressure of at least 125 psi. Copper tubing may be of the standard K or L grade.

Pipe joints may be screwed, flanged, welded, brazed, or soldered, but solder, if used, must have a melting point of at least 1000 F.

If working pressures exceed 125 psig, then 250-lb pipe and fittings must be used, but cast-iron should not be used at any pressure.

Filling Densities. Storage tanks are rated on their water capacities. For safety reasons, these tanks must not be filled beyond a certain percentage which is specified by the National Board of Fire Underwriters. Typical filling densities are shown in Table 14.15.

Liquid Transfer Operations. When transferring liquefied petroleum gas from the delivery container to the storage tank, two methods are recommended. First, the liquid may be pumped, using a pump approved by the Underwriters' Laboratories which may be driven by the truck motor, if the delivery is being made by truck. Second, transfer may be accomplished by a compressor, also approved by the Underwriters' Laboratories. In the compressor method, the compressor normally takes its suction from the vapor space of the storage tank, and discharges into the vapor space of the container being emptied. The pressure so built up in the delivery container causes the liquid to flow into the storage tank where the pressure is kept low by the compressor.

At times of cold weather, the vapor pressure of the liquid in the storage tank may be so low that the system will not function as described, the compressor being unable to harvest enough vapor to build up a pressure in the delivery container. When this happens, the compressor is permitted to take its suction from the air, but the air so compressed must be dehydrated before it is discharged into the container being emptied.

TABLE 14.15. FILLING DENSITIES FOR LP GAS TANKS, PER CENT

Specific gravity of gas at 60 F (air = 1)	Aboveground tanks, up to 1,200 gal water capacity	Over 1,200 gal water capacity	Underground, all capacities
0.473–0.480	38	41	42
0.496–0.503	41	44	45
0.520–0.527	44	47	48
0.545–0.552	47	50	51
0.561–0.568	49	52	53
0.577–0.584	51	54	55
0.593–0.600	53	56	57
0.609–0.617	55	58	59
0.627–0.634	57	60	61

Cold weather also may affect the satisfactory operation of a liquefied petroleum system in an adverse manner, unless the system is equipped at the tank with some form of vaporizer. Normally, a slight reduction of pressure over the surface of the liquid in the storage tank causes some of the liquid to vaporize. Domestic systems generally operate on this principle, the heat of the ambient air being adequate to provide the 170 Btu per lb needed for vaporization. Industrial systems, critically dependent on a continuous flow of gas, dare not risk interruptions of the supply due to cold weather; consequently steam or hot water coils may be built into the storage tank, and activated automatically when needed.

Cylinder Systems. The storage tanks previously discussed have been of the permanent variety, that is, they stay where they are placed for their entire lives, or at least are so intended.

We shall now look at the cylinder systems. In this arrangement, vertical cylinders are delivered to a user, hooked up to the piping distribution system, and then, when empty, replaced by filled cylinders. This cycle of replacement goes on so long as the supplier and his customer continue friendly relations.

These cylinders, which particularly carry the "bottle gas" designation to most users, must conform in their construction to all the basic rules for unfired pressure vessels as set up by the American Society of Mechanical Engineers. In addition, since they are portable, they must bear the approval of the Interstate Commerce Commission.

These containers must not be placed underground, and the safety relief valves must be located at least 5 ft away, horizontally, from any building opening which is present *below* the level of the discharge. The level stipulation recognizes the fact that these gases are of greater density than air, and consequently sink to the ground when vented.

The piping of the containers must be so arranged that the replacement of emptied cylinders may be accomplished without interrupting the flow of the gas.

Sizing a Gas System. The procedures for sizing a gas system are fundamentally uniform for all of the common gaseous fuels. For that reason, simply by correcting for density in the formulas, we can size satisfactorily for natural gas, artificial gas, or liquefied petroleum gases.

The formulas are shaped to meet two conditions of gas handling: (1) gas at low pressure, generally considered to be less than 1 psig, and (2) gas at high pressure, or anything higher than 1 psig.

Various investigators have produced various formulas, and in order to acquaint ourselves with the available choice, here are several that are favored:

1. *The Spitzglass formula* for low-pressure distribution is

$$h = \left(\frac{Q^2SL}{3{,}550^2}\right)\left(\frac{d^3}{1 + \frac{3.6}{d} + 0.03d}\right)$$

where h = pressure drop, in. of water
Q = gas flow, cu ft per hr
S = specific gravity of the gas (air = 1.00)
L = equivalent length of pipe, ft
d = internal diameter of pipe, in.
P = pressure drop, psi
a = mean absolute pressure in the pipe, psi

2. *The Spitzglass formula* for high-pressure distribution is

$$Q = 4{,}830\left[\frac{Pad^5}{SL\left(1 + \frac{3.6}{d} + 0.03d\right)}\right]^{\frac{1}{2}}$$

3. *The Pole formula* for low-pressure flow is

$$Q = 1{,}350d^2\sqrt{\frac{pd}{0.33 \times sL}}$$

The letters in this formula have the same meanings as given above for the Spitzglass formulas, plus p = pressure drop, in. W.G.

4. *The Weymouth formula* for high-pressure flow is commonly used, particularly for long pipe-line problems, and the results tend to be notably accurate. Here is the formula:

$$Q = 433.5 \frac{T_0}{P_0} \sqrt{\frac{(P_1^2 - P_2^2)d^{5.33}}{GTL}}$$

where Q = cu ft of gas per 24 hr, measured at the temperature and pressure at the point of delivery
T_0 = absolute temperature of gas, at delivery end
P_0 = absolute pressure, psig, at delivery end
P_1 = absolute pressure, psig, at initial end of line
P_2 = absolute pressure, psig, at terminal end of line
d = internal diameter of pipe, in.
G = Specific gravity of gas
T = absolute temperature of gas as it flows through pipe
L = length of pipe line, miles

Since this formula was developed primarily for large gas-transmission lines, we must remember that the temperatures at the point of delivery and the temperatures at the terminal end of the line can be different, as trunk lines commonly deliver gas at intermediate stations. For most of our short run heating installations, it will be safe to assume that the temperature T of the gas flowing through the pipe is identical with the temperature T_0, at the delivery end. In other words, the temperature will be relatively constant. The effect of temperature changes, of course, is to modify the volume of the gas, the volume varying as the square root of $1/T$, with the T being the absolute temperature.

5. *The Oliphant formula,* also widely favored for natural gas transmission, is

$$Q = 1{,}008 \times d^{2.542} \sqrt{\frac{P_1^2 - P_2^2}{L}}$$

This formula is based on a specific gravity of 0.6.

The results from the solution of any one of these formulas will be well within the range of practical accuracy; that is, pipe lines so designed will work. However, the repeated lengthy exercises in arithmetic become quite tedious, and consequently most designers prefer to size their gas lines by means of a standard nomograph.

These nomographs are available from the same sources that issue the oil-piping charts. Figures 14.12 and 14.13 are typical nomographic

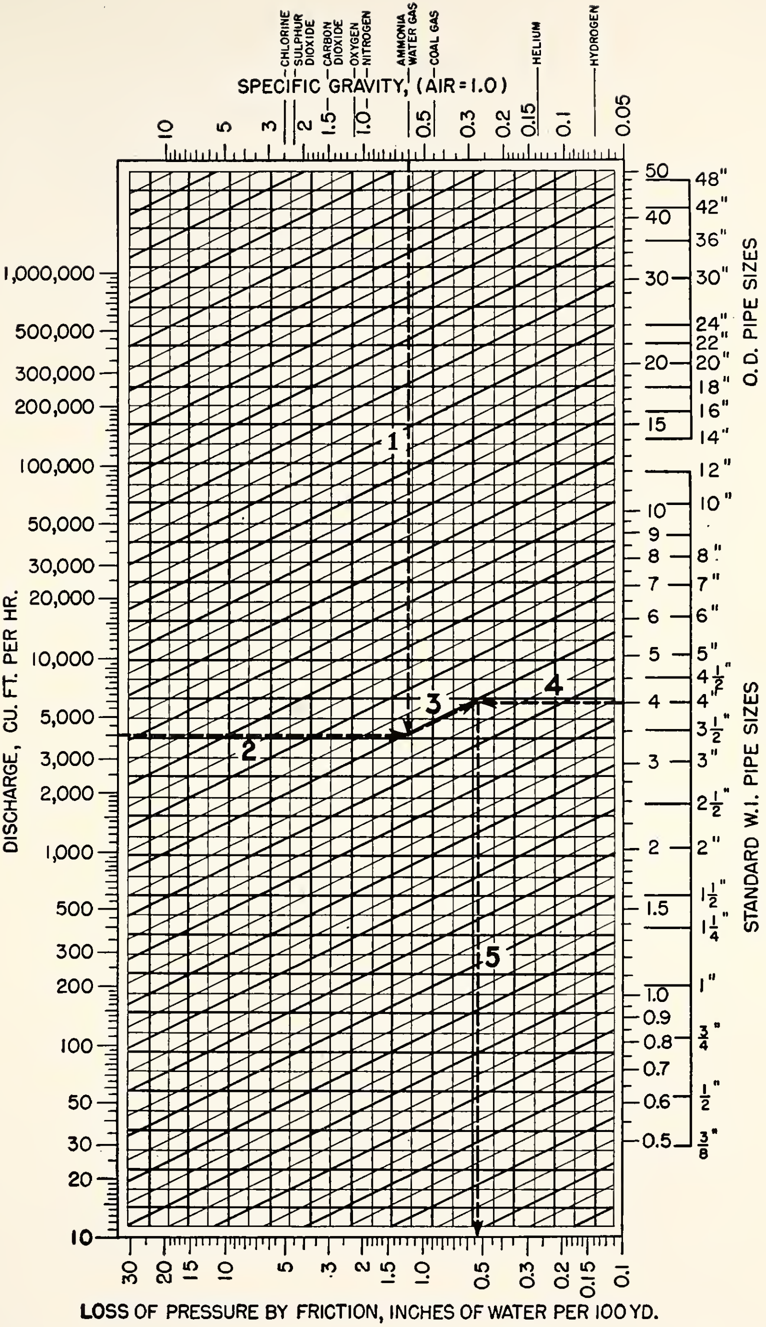


Fig. 14.12. Nomograph for low-pressure gas piping. *Example:* 4,000 cu ft per hr of gas of specific gravity of 0.6 flowing through a 4-in. line, to find the pressure drop. Proceed by steps 1, 2, 3, 4, 5 as indicated by the dotted lines, reading the pressure drop as 0.53 in. of water per 100 yards. *Answer:* 1 in. water = 0.036 lb per sq in. (*Courtesy of Crane Company.*)

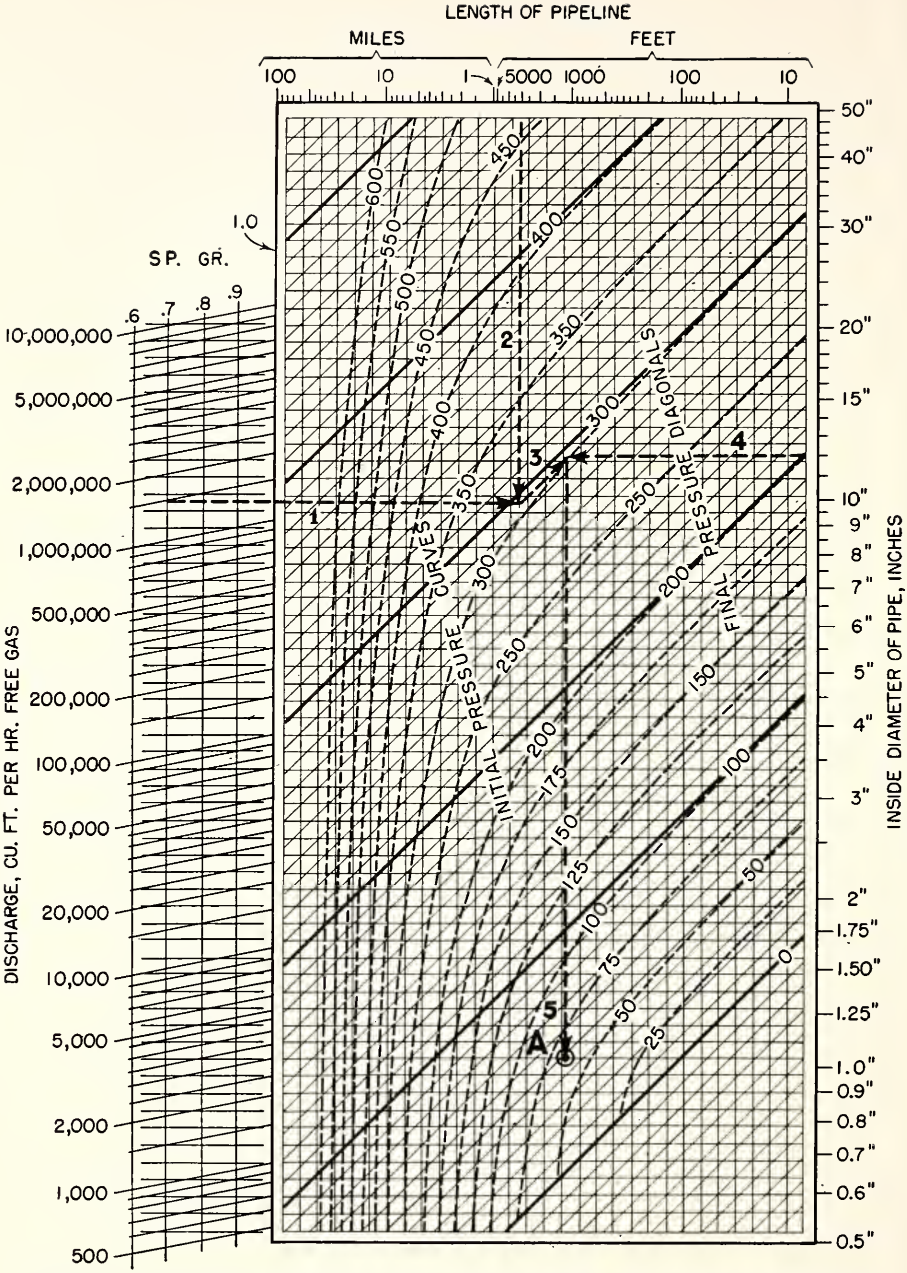


Fig. 14.13. Nomograph for sizing high-pressure gas piping. *Example:* A 12-in. pipe line 3,000 ft long discharges 1,500,000 cu ft per hr of gas having a specific gravity of 0.7. If initial pressure is 65 lb per sq in. absolute what is the final pressure? Follow steps 1, 2, 3, 4 and 5 shown on diagram, thus, enter top at 3,000 ft then down to 1,600,000 cu ft per hr then down to 65 lb initial pressure curve and read answer 48 lb on intersecting final pressure diagonal. (*Courtesy of Crane Company.*)

arrangements, and will be found suitable for the solution of both high-pressure and low-pressure problems.

Step-by-step Procedure. If we are running a gas line to a single heater, the problem is quite simple, even if we choose to use a mathematical equation rather than the nomograph. For example, suppose that we must deliver 500 cu ft of gas an hour to a heater that is 30 ft from the gas service connection. For this demonstration, we will assume that we have a pressure of 20 in. W.G. at the public utility service connection with gas of 0.6 specific gravity, and furthermore that we can stand a pressure drop up to the burner of 15 in. W.G.

TABLE 14.16. EQUIVALENT LENGTH OF VALVES AND FITTINGS FOR GAS PIPING SYSTEMS, FEET*

Pipe diam., in.	Globe valve	Plug cock	Elbows	Tees (run)
½	3	3	1	1
¾	4	3	2	1
1	5	4	2	1
1¼	7	6	3	1½
1½	8	6	3	2
2	11	9	4	2
2½	14	11	5	3
3	18	13½	6	3
4	24	18	8	4
6	39	30	13	7
8	54	41	18	9
10	70	53	24	12
12	88	66	30	15

* *Note:* Head loss through fittings is reported by different investigators to have different values. This table is offered by the author as having been the result of a weighing and an evaluating of the reports by various authorities. No figure is claimed to be exact, and at best should be considered an approximation. Designers having information on pressure drops from sources in which they have confidence should make use of those sources. This table is intended as a guide in the absence of exact data.

Since 500 cu ft per hr is not a great volume of gas, and we are permitted a substantial friction loss, we will endeavor to keep down the cost of the piping by checking the loss in a ¾-in. pipe.

Using the Pole formula, which is comparatively short and simple, we have

$$500 = 1{,}350 \times 0.75^2 \sqrt{\frac{p \times 0.75}{0.33 \times 0.6 \times 30}}$$

$$p = 3.47 \text{ in. W.G.}$$

Obviously, a $\frac{3}{4}$-in. pipe is larger than we need. In a situation like this we will try again, using a $\frac{1}{2}$-in. pipe, and the equation is

$$500 = 1{,}350 \times 0.5^2 \sqrt{\frac{p \times 0.5}{0.33 \times 0.6 \times 30}}$$

$$p = 26 \text{ in. W.G.}$$

This loss is more than we can accept; consequently it looks as if $\frac{3}{4}$-in. pipe will be our choice.

Sometimes, the heating designer is faced with the problem of running gas piping to a dozen or more direct-fired heaters, to gas-steam radiators, and to gas-fired kitchen or process equipment. In these cases, the designer must first lay out an arrangement of the piping, showing the loads on each branch, the cubic feet per hour of gas required, and the length in feet or equivalent feet of each trunk main and branch. If distances are in feet, each valve and fitting should be shown, in order that the equivalent length of the piping may be calculated. Figure 14.14 is such a diagram, and it clarifies the problem considerably.

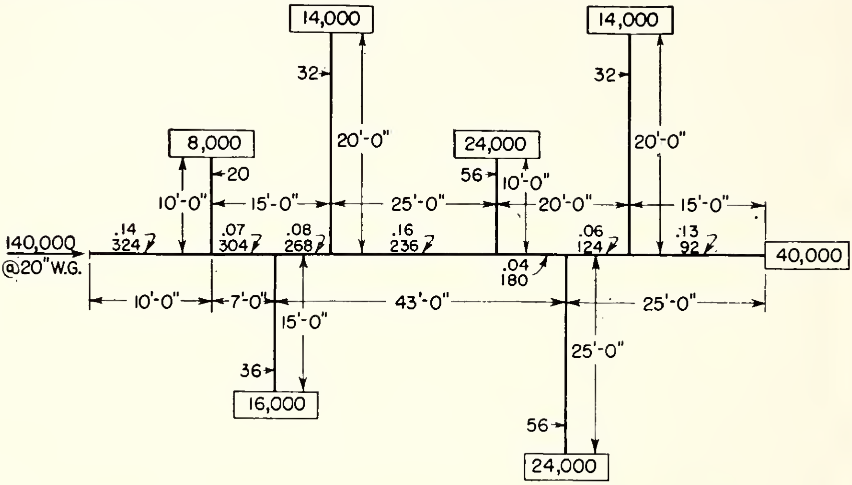


Fig. 14.14. Diagram for sizing gas piping by showing sectional loads.

By starting with an input pressure, and knowing the minimum pressure that is necessary for the proper functioning of each unit, we can set down on the diagram the acceptable pressure drops for any given length of pipe.

As we are dealing in volumes, we must also convert the various loads in Btu to cubic feet of gas per hour, using the heating value of whatever gas we happen to be using for the job. It is wise also to calculate the cubic feet per hour on a burning efficiency of 80 per cent. That is, if we have a gas of 560 Btu per cu ft, only 560 × 0.80 or 448 Btu will be of use, the balance being dissipated in radiation, chimney loss, and so on.

With all this necessary information set down on the piping arrangement sketch, we are ready to proceed with the sizing. Each section must be sized separately, since we plan not only to keep within the allowable pressure drops but also to hold down the piping cost as much as possible.

Whether these sections are sized by the use of a formula or by nomograph is up to the individual. The latter is the faster, and can be more accurate, the human capacity for making mistakes being what it is; but here and there we will find designers to whom the long way around is always the most satisfactory way home.

Some Thoughts on Safety. Gas fuel is not only flammable but it is also explosive. Keeping this fact in mind, the designer should be doubly careful to design against all possible leaks and interruptions to normal combustion.

A burner supply valve that continues to pass gas after it has been apparently closed is a potential angel of death. This is particularly true in large industrial heaters that may be shut down manually at night and relighted manually in the morning. A torch pushed into a furnace that has filled with gas during the night becomes almost as destructive as dynamite.

Before lighting a gas-fired unit of any sort, it should be vented and ventilated in every possible way, even if the pilot failed only 10 sec before.

Figure 6.28, Chap. 6, illustrates a safety arrangement for industrial gas burners that has been developed by insurance casualty engineers. By piping the cocks as shown, a very tiny leak is sufficient to keep the system inoperative, and will continue to do so until repairs have been made.

Questions

1. Name eight major forms of coal.
2. What is the outstanding characteristic of each form?
3. Where does each form find its greatest field of usefulness?
4. Would you store a winter's supply of coal in the open? If not, why not?
5. How do you check a coal pile for temperature?
6. How large a bin must you have to store 20 tons of coke?
7. Describe three methods for removing ash from industrial boilers.
8. What is hogged fuel, and where is it used?
9. Describe a sawdust burner suitable for a domestic heater.
10. Name seven characteristics that are measured and evaluated in classifying fuel oil.
11. How does kerosene differ from the conventional fuel oils?

12. What is the relationship, if any, between the Baumé and API scales?

13. Is there a limit on the quantity of fuel oil that may be stored in a building? What is the limit?

14. What is the difference between a positive displacement pump and a centrifugal pump?

15. What is an oil-tank "dike" and why is it used?

16. Differentiate between kinematic viscosity and absolute viscosity.

17. What useful purpose in piping design is served by the Reynolds numbers?

18. What is a nomograph? What is its value, if any?

19. Name the three major gaseous fuels.

20. What is the cardinal rule to be observed when lighting a gas burner in a gas-fired unit of equipment?

CHAPTER 15

COMBUSTION

Principles

A working knowledge of what happens to fuel in a furnace or boiler is essential if we are to design adequately our chimneys and stacks, provide equipment that will operate efficiently, and solve service questions with understanding.

We all know that a fire will not burn without air, but sometimes we tend to overlook the fact that different kinds of fires require different quantities of air. A change in fuel for any given furnace and chimney system potentially is able to transform a satisfactory installation into a troublemaker simply because the existing air and gas facilities are unsuited to the new fuel.

Let us examine some different types of fuel with the objective of discovering how much air each requires, and why.

Air for Burning Coal. Coal contains three burnable elements; carbon, hydrogen, and sulphur. The reaction of the carbon with the oxygen in the air (if the temperature is high enough to produce ignition) forms carbon monoxide (CO) and carbon dioxide (CO_2). The carbon monoxide results generally if the air is skimpy; and we do not like carbon monoxide, not only because it is a suffocating gas, but because when carbon burns into CO, it does not yield nearly so much heat. Our coal dollars therefore go up the stack rather uselessly.

When the carbon in coal burns to carbon dioxide, the heat released in the reaction is 14,540 Btu for every pound of carbon; but if it burns to CO, the Btu shrink to 4380. The answer of course is to give our carbon enough air.

Now here is something to remember: The actual quantity of air we must make available is always somewhat greater than the theoretical demand, since some of the air introduced into a furnace stratifies and slips through without reacting with its assigned partner, carbon. We must therefore give the forsaken carbon a second and even a third chance, and the air for these extra chances is called "excess." Excess air alone, however, does not guarantee complete combustion. Contact between the carbon and air is essential, and if we just introduce a deluge of air without making sure that it meets the carbon in intimate embrace, our efforts to produce CO_2 are wasted.

The second combustible element in coal is hydrogen. When hydrogen burns, the reaction produces water in the midst of the flame, but the heat release is tremendous, 62,000 Btu for every pound of hydrogen. With a heat like this, the water has no chance to function as water, but instantly becomes superheated steam, and as such passes out the stack.

The heat which transforms the water into steam is lost to us, and represents a substantial debit if the percentage of hydrogen in the coal is high.

The third heatmaker in coal is sulphur. This yellow material burns into sulphur dioxide, and in the process yields 4050 Btu for each pound of sulphur that we have to start with.

Generally the weight of sulphur is quite low, perhaps only 2 or 3 per cent, and although SO_2 (sulphur dioxide) is quite poisonous, there is seldom enough going out the stack to be of any moment to anybody. Sulphur has some other undesirable characteristics, which are described in Chap. 14.

Each of these three combustibles, carbon, hydrogen, and sulphur, has its own peculiar requirements for air. These requirements for theoretically perfect combustion, which means no carbon monoxide in the case of carbon, are as follows:

1 lb carbon.............	11.52 lb air
1 lb hydrogen..........	34.56 lb air
1 lb sulphur............	4.32 lb air

Now let us see what this means in the case of each pound of coal we contemplate burning in a projected heating plant. Incidentally whether we throw the coal into the furnace by hand, slide it in on a stoker, or grind it up and blow it in, the requirements for perfect combustion, being set by a law of nature, are exactly the same. Only the excess air may fluctuate, for control of the excess is a matter of human judgment and experience.

We begin by obtaining an analysis of the coal, preferably an ultimate analysis, although a proximate one may be used with some modifications which we shall examine a little later. A typical analysis of bituminous coal, showing the per cent by weight and the weight of air for combustion for each constituent, is given in Table 15.1.

There is a refinement to this gross figure that we are entitled to employ; it involves taking advantage of the oxygen which appears in the analysis of the coal. For all practical purposes, air is composed of only two elements, oxygen and nitrogen, and it is only the oxygen that reacts with our combustible materials; the nitrogen performs no useful function, except possibly to prevent the oxygen from becoming lonely in the vast spaces of the world. Each pound of air therefore is made up of 23.15 per cent oxygen (3.704 oz) and 76.85 per cent nitrogen (12.296 oz).

Since our coal analysis shows that the coal already contains 4.5 per cent oxygen and needs merely the incidence of an ignition temperature to unite with its neighboring carbon to form CO_2, we can hold back the quantity of air that would be needed otherwise to deliver that 4.5 per cent of oxygen to the furnace. Very well, 4.5 per cent of a pound of oxygen travels around as 23.15 per cent of 0.194 lb of air, as we discover by 0.045/0.2315 = 0.194. If this way of determining the equivalent air for 4.5 per cent oxygen appears obscure, just convert everything into ounces and all will be clear.

TABLE 15.1. TYPICAL ANALYSIS OF BITUMINOUS COAL

Constituent	*Per Cent by Weight*
Carbon	80.0
Hydrogen	5.0
Oxygen	4.5
Sulphur	2.0
Nitrogen	2.0
Ash	6.5
Total	100.0

AIR NEEDED FOR COMBUSTION

For the carbon	80% of 11.52 lb	=	9.22 lb
For the hydrogen	5% of 34.56 lb	=	1.73
For the sulphur	2% of 4.32 lb	=	0.09
Gross total			11.04 lb

Anyway, we can subtract 0.194 lb of air from 11.04, and we find that the answer is 10.846 lb.

The next question of course is: How much excess air shall we figure on? For hand firing and in small domestic installations where no attempt is made as a rule to operate efficiently, anywhere from 50 to a 100 per cent. Good operation of oil burners occurs with 30 per cent, while stoker practice may average 40 to 50 per cent excess air.

Proximate Analysis. Many coal dealers have no idea of how their wares look when dissected by an ultimate analysis. To obtain an ultimate analysis, they must go back in most cases to the mine. A proximate analysis is usually available, however, although we must be careful to verify it as pertaining to the coal with which we are working. Dealers obtain their supplies from one or several mines, and the proximate analysis of a West Virginia coal, if handed to us casually whereas we actually are buying Illinois coal, can lead us to false estimates.

Proximate analyses itemize the following factors: (1) fixed carbon, (2) volatile substances; (3) sulphur, (4) ash and moisture, and (5) heating value, Btu per pound.

Now everything we need is here, but a substantial percentage of carbon and all the hydrogen are jumbled together as volatile substances. Somehow we must estimate the proportions of each.

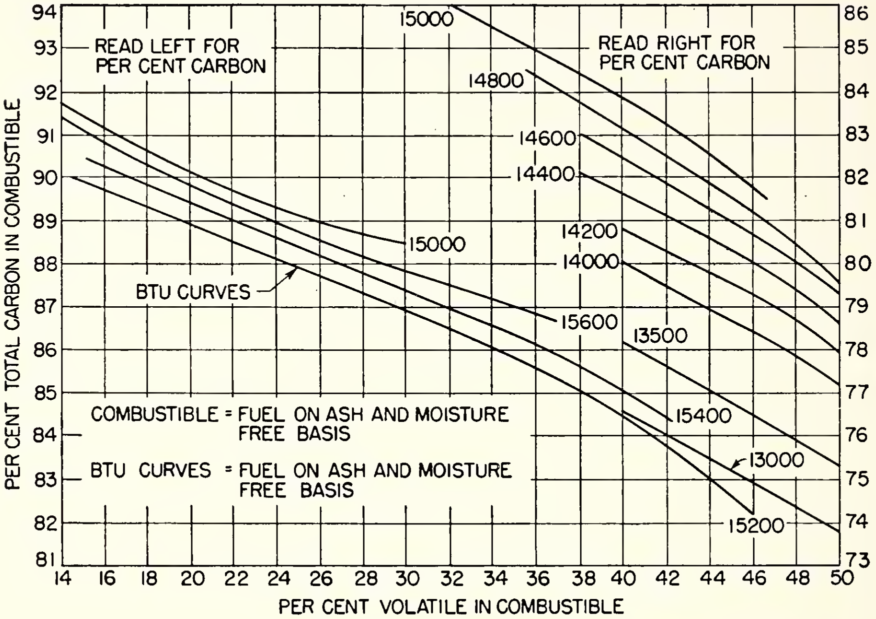


Fig. 15.1. Fuel analysis curves. *(Developed from data made available by the American Boiler Manufacturers' Association.)*

One way is to examine the records of the field from which the coal is mined. The U.S. Bureau of Mines can tell us, too. Both of these means involve time. A faster method is to use the curves developed by the American Boiler Manufacturers' Association, which are reproduced here as Figs. 15.1 and 15.2. Suppose we assume a proximate analysis, as in Table 15.2, and try out these curves for practice. The values given

Table 15.2. A Proximate Analysis of Coal

Constituent	*Per Cent*
Fixed carbon	57.50
Volatile	36.40
Sulphur	0.60
Ash and moisture	5.50
Total	100.00

describe the character of the coal as it is dumped on our coal pile; consequently our first step is to reevaluate these percentages on the ash- and moisture-free basis needed for using the curves. The procedure is a simple act of arithmetic in which we add together all the percentages of the things we want, and then use this total as the divisor for each of the units that went into the total:

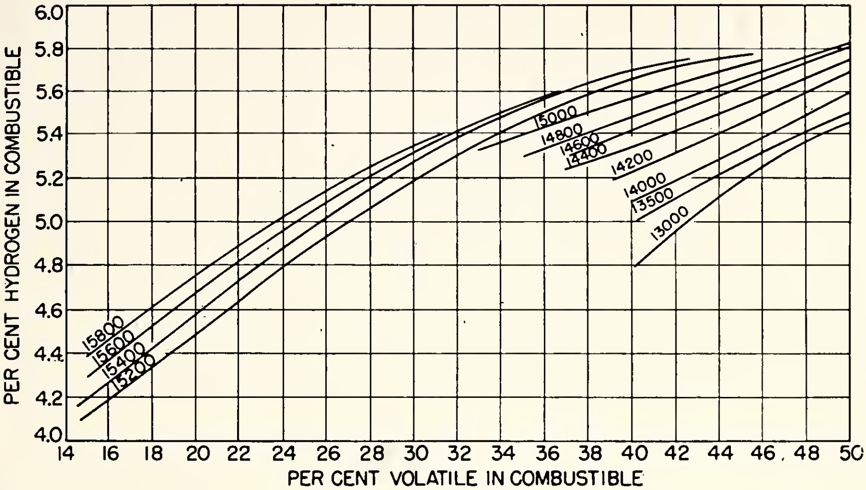


Fig. 15.2. Relative hydrogen curves. *(Developed from data made available by the American Boiler Manufacturers' Association.)*

Fixed carbon 57.50 + volatile 36.40 + sulphur 0.6 = 94.50

Next,

$$\frac{57.50}{94.50} = 60.85\%; \qquad \frac{36.40}{94.50} = 38.52\%; \qquad \frac{0.6}{94.50} = 0.63$$

The same procedure is applied to the heating value of the delivered coal, which we shall say is reported by the dealer as being 14,100 Btu per lb; 14,100/0.945 = 14,920 Btu on the ash- and moisture-free basis.

Now we are ready to refer to the first curve (Fig. 15.1) with the volatile percentage of 38.52, and reading upward to the 14,920 curve (which happens to fall between the 14,800 and 15,000 curves and is not shown), the percentage of carbon in the fuel appears in the right-hand column as approximately 83.8.

The same procedure is now followed on the hydrogen curves (Fig. 15.2). The answer is 5.5 per cent hydrogen.

One step remains. We must return our percentages to the "as received" basis in order to estimate the air needed, and the heating value, for each pound of coal as we buy it. Once more a little arithmetic becomes our ally. This time we multiply the carbon and hydrogen, newly taken from the curves, by 94.5 per cent. The answers are carbon, 79.19 per cent; hydrogen, 5.20 per cent; sulphur as we first had it, 0.6 per cent. If we add all these together, and the ash and moisture too, we have a total of 90.49 per cent. The difference between this value and 100 per cent represents the oxygen and nitrogen in the fuel, and we must estimate these individually. Nitrogen generally varies between 1.5 and 2.0 per cent; thus if we assign it a value of 1.75 per cent, which is halfway between these limits, the oxygen becomes 7.76 per cent.

Obviously this method of converting a proximate analysis to an ultimate one is not precisely accurate. The error, however, is too small to affect our practical activities; we can use it without trembling for determining the air required for combustion and for calculating the nature of the gases that go up the chimney. In short, despite its lack of absolute accuracy, this method of conversion is a good one to know.

Air for Burning Oil. Since the burnable elements in fuel oil consist of carbon, hydrogen, and sulphur, the procedure for finding the air needed for combustion is precisely the procedure we used for coal. We shall be impressed, however, by the relatively high percentage of hydrogen present in the oil. It is this hydrogen that contributes the major portion of the 4000 to 5000 Btu per lb heat advantage enjoyed by oil, when compared with coal. Table 15.3 shows a typical analysis of fuel oil, from which we can see what percentage of air is needed for combustion.

TABLE 15.3. A TYPICAL ANALYSIS OF FUEL OIL

Constituent	*Per Cent*
Carbon	84.00
Hydrogen	13.00
Oxygen	1.10
Sulphur	0.40
Nitrogen	1.50
Total	100.00

AIR NEEDED FOR COMBUSTION

Carbon	0.84	×	11.52	=	9.68 lb of air
Hydrogen	0.13	×	34.56	=	4.49
Sulphur	0.004	×	4.32	=	0.02
Gross total					14.19
Credit oxygen equivalent					0.05
Net total					14.14 lb of air

Compare this figure of 14.14 lb of air per pound of oil with the 10.849 lb we previously computed for coal; the difference is caused mostly by the hydrogen preponderance in the oil.

Excess air is applied more thoroughly with oil than with coal, since the atomization of the oil by the burner is really an air and oil mixing process. For estimating, 15 to 20 per cent is a fair average; consequently if we allow the latter amount, which means 20 per cent of 14.14 lb, or 2.83, lb, we reach a practical 16.97 lb of air to be furnished for burning each pound of oil.

In domestic installations, the air for oil burners will vary from a little less to approximately equal that found with coal, due mainly to the high percentages of excess resulting from a habitual lack of control in hand-fired furnaces. Stokers are more efficient.

Sometimes an oil burner is substituted for a grate in order to overcome poor stack and combustion conditions; in most cases the simple addition of a small, forced draft fan would produce equal satisfaction.

Air for Artificial Gas. When we obtain an analysis of *gas fuel*, we are almost certain to receive it on the basis of percentages by *volume*. Suppose, for example, that our analysis is the one given in Table 15.4.

TABLE 15.4. AN ANALYSIS OF GAS FUEL

Constituent	*Per Cent by Volume*
Carbon dioxide	1.00
Carbon monoxide	5.50
Methane	28.00
Hydrogen	53.00
Nitrogen	12.50
Total	100.00

In order to determine the air needed for burning this sort of mixture, we must convert the volume percentages to weight percentages. The process (see Table 15.5) is one of simple arithmetic, merely a first step of multiplication, followed by addition and division, such as we already have experienced in converting proximate to ultimate analyses for coal.

TABLE 15.5. CONVERSION OF VOLUME PERCENTAGES TO WEIGHT IN POUNDS FOR GAS FUEL

Gas	Weight, lb/cu ft	% present in analysis per cu ft	Weight, lb
Carbon dioxide	0.12341	1.00	0.00123
Carbon monoxide	0.07806	5.50	0.00429
Methane	0.04500	28.00	0.01260
Hydrogen	0.00562	53.00	0.00298
Nitrogen	0.07807	12.50	0.00976
Total weight	....	.....	0.03086

Now if we divide each of these weights in pounds by the total weight of 0.03086 lb per cu ft, we obtain the percentages we are seeking. The answers are given in Table 15.6.

Table 15.6. Weight Percentages for Gas Fuel

Constituent	*Per Cent by Weight*
Carbon dioxide	4.00
Carbon monoxide	13.90
Methane	40.83
Hydrogen	9.63
Nitrogen	31.64

With these weights known, we can proceed to determine the air weights for each burnable constituent. Hydrogen already is familiar to us, but the idea of carbon monoxide being combustible may be new, and certainly we have not met methane, the "marsh gas," in our previous calculations. The procedure, however, is a familiar one, and we can set down our findings in Table 15.7.

Table 15.7. Determination of Air Weights for Each Burnable Constituent

Combustible	% present	Air for 1 lb of combustible	Air for % present, lb
Carbon monoxide	13.90	2.46	0.342
Methane	40.83	17.28	7.055
Hydrogen	9.63	34.56	3.328
Total	.....	.....	10.725

For each *cubic foot* of gas, therefore, we shall have to supply, theoretically, 10.725 lb of air for combustion. This air can be changed into cubic feet if we know the temperature at which it is being supplied to the gas burners. For example, if the air comes from an open room in which the temperature is 80 F, we must corral 13.599 cu ft of it before we have a pound. For the 10.725 lb of our problem, the total becomes, therefore, 145.85 cu ft.

The volume of air at other temperatures is shown in Table 2.7.

Manufactured gases differ in various localities. The analysis used in this example is a coal gas produced by a coke oven. In some cities, only a small part, or none, of the gas distributed through the mains is strictly a coal gas; instead we shall find a composite gas that has drawn on the oil refinery and the carbureted water gas machines for its substance. The water gas, which is formed by the dissociation of steam when passed through a bed of glowing coke, is somewhat lean in heat value, and in order to meet city requirements of, say, 525 to 550 Btu per cu ft it must be enriched. Commonly, the enriching procedure is to spray a good grade of oil ("gas oil" it is called) over hot checkered brick in a chamber where the water gas is already hurrying its lean bulk through. The gas oil gasifies from the heat of the brick and joins the water gas in leaving that hot spot as promptly as possible. Together, the gases pass into another chamber, similarly furnished with hot checkered brick, and there they are merged by the heat to form the gas that fries domestic eggs and provides fuel to industry.

A water gas plant commonly consists of three chambers. In the first, by name the generator, the steam meets the coke and becomes a pale lean gas. The second or middle chamber, in which the gas oil becomes oil gas, is called the "carburetter." The final merging takes place in the "superheater."

The process is an intermittent one, since the brick cools and the coke in the generator must be replenished and whipped up to high heat by a blast of forced draft before the steam can be introduced.

Regardless of the analysis of the gas, that is, whether it is coal gas, water gas, or oil refinery gas, the procedure for estimating the air required for combustion follows the steps we have already taken.

Excess air is comparatively low for gas fuel. In most cases, from 12 to 15 per cent is all we need count on. Thus for each cubic foot of the gas in our example, our estimate of air required will be 10.725 lb plus 15 per cent, or 12.334 lb.

Air for Natural Gas. We handle combustion questions for natural gas exactly as we did for manufactured gas. The first step is to obtain an analysis of the gas.

The combustible elements in natural gas are somewhat more numerous, which imposes on us a larger burden of arithmetic, but that is all. In addition to the carbon monoxide, hydrogen, and methane of a coal gas, we meet ethylene and hydrogen sulphide, a pair of newcomers that also will burn and yield some heat. Also, we are likely to discover some oxygen in the analysis, and therefore are entitled to credit our computations with an equivalent amount of air, exactly as we did in the coal case. An analysis from an Ohio field (Table 15.8) shows that natural gas, while it contains a wide variety of constituents, is predominantly methane.

TABLE 15.8. AN ANALYSIS OF NATURAL GAS FROM AN OHIO FIELD

Constituent	*Per Cent by Volume*
Carbon monoxide	0.45
Hydrogen	1.82
Methane	93.33
Ethylene	0.25
Hydrogen sulphide	0.18
Oxygen	0.35
Carbon dioxide	0.22
Nitrogen	3.40

Without detailing the steps of changing these combustibles into percentages by weight, an analysis such as this will require 15.91 lb of air for each cubic foot of the gas. This is much greater than we found needed by manufactured coal gas, the increase being due to the preponderance of methane.

Excess air, however, remains low. From 12 to 15 per cent is the usual range observed when combustion is good. An allowance of 15 per cent for our estimates produces a total of 18.30 lb of combustion air for each pound of this natural gas.

Obviously, a combustion system set up for burning artificial gas may not operate as successfully with natural gas. This is one of the possibilities we must consider when designing a system that is intended to operate with either fuel.

Air for Burning Wood. Since wood in a manner of speaking is coal in the raw, we are not surprised to discover that carbon and hydrogen are the burnable constituents.

Wood comes to the fire more often than not heavily charged with moisture. Evaporation of this water debits the wood's production of heat by approximately 971 Btu per lb, which is the heat of vaporization, and to this loss there accrues also all those Btu used in superheating the newly formed vapor to the exit temperature of the furnace.

To estimate the air needed for burning wood, we must first place the analysis on a moisture-free basis. Suppose that we take a 2-lb chunk in which the moisture represents 50 per cent of the weight. After drying the sample, which now weighs only 1 lb, the ultimate analysis looks like Table 15.9.

TABLE 15.9. ULTIMATE ANALYSIS OF 1-POUND CHUNK OF DRIED WOOD

Constituent	*Per Cent*
Carbon	50.31
Hydrogen	6.20
Oxygen	43.08
Nitrogen	0.04
Ash	0.37

By proceeding in the usual way of multiplying the air required for burning carbon and hydrogen, with the percentage of each present, we find that the gross weight of air for burning the wood is 7.939 lb. Now we credit the air equivalent of that big percentage of oxygen. It amounts to 1.862 lb, leaving a net requirement of 6.077 lb.

Apparently wood will burn with a very little air—theoretically. Actually, a considerable excess must be used, from 50 to 100 per cent. The main reason for this is the difficulty in achieving efficient contact between the air and wood. The usual wood fire of both domestic and industrial furnaces feasts on whatever is handy, shavings, chips, chunks, or some roughly sized pieces known as "hogged" fuel. These materials when thrown together on a grate are honeycombed with alleyways and corridors through which the air takes a short cut to the stack. To complicate our contacting problem further, charring the surfaces of some woods tends to insulate the combustible underneath; consequently air may pass right over the potential fuel without any reaction taking place. Every owner of a home fireplace has watched this phenomenon; nothing less than a chipping away of the charred surface seems to rekindle a smouldering log.

We can estimate 100 per cent excess air for wood burning in most of the stoves and heaters intended for using this fuel. Considerable work has been done in the Pacific Northwest to design an efficient domestic burner. Industrial plants that use wood refuse for both steam and heating normally feed the material to the grate through a loading manhole in the top of a Dutch oven furnace. Air enters the fire from an open door under the grate, and the quantity is as much as the draft will handle.

Air for Burning Liquefied Petroleum Fuels. These "bottled gases" are by-products of the more important gasoline cracking processes. Two names commonly appear in the commercial fuel list, namely, propane and butane. Each is a combination of carbon and hydrogen, the former bearing the chemical identification of C_3H_8, the latter C_4H_{10}. The air required to burn a pound of propane is 15.66 lb; for butane, 15.445 lb. Excess will be low; 10 per cent is generally a safe estimate.

Air for Burning Bagasse. Since bagasse is merely sugar-cane stalk, the heating designer is unlikely to face many problems with this fuel, for sugar cane grows only in those areas of the world where heating for comfort is never, or rarely, necessary. It is, however, an industrial fuel of some importance.

TABLE 15.10. A TYPICAL ANALYSIS OF BAGASSE

Constituent	*Per Cent*
Carbon	47.00
Hydrogen	6.00
Nitrogen Oxygen	40.00
Ash	7.00
Moisture, as received	50.00

Since it is a vegetable fiber, its characteristics are similar to those of wood: it is normally heavy with moisture, dirty, and fouled with impurities such as sand and mud. A typical analysis of bagasse is given in Table 15.10.

If we must, we will proceed to estimate the air for combustion in the manner we employed when considering wood, and add 100 per cent for excess.

The burning of bagasse has been a fascinating problem to many engineers. There are at least a dozen designs of furnace, including flat grates, sloping grates, Dutch ovens, and step grates.

What Air Does to Fuel

Combustion is a chemical reaction that takes place when the temperature of a material exceeds its particular ignition temperature. Temperature is like a catalyst: mix combustibles and air under ordinary ambient conditions and nothing happens; raise the temperature sufficiently and the combustible and air unite with fireworks.

The reason that carbon requires a certain amount of air, while hydrogen, sulphur, and carbon monoxide demand something else, is discovered in the chemistry of the reaction. Suppose that we break down the union of carbon and oxygen and see why 11.52 lb of air are essential for this purpose.

The Chemistry of Burning Carbon. We start by considering the relationship of molecules and atoms. A molecule is representative of a complete substance, while an atom is one of the few or many constituents, often dissimilar, that are combined to form the molecule. For example, if we view an apple pie as a molecule, then the apple in it may be termed one atom, and the dough another atom.

Atoms are classified and weighed, their relative weights being referred to hydrogen. Table 15.11 shows established atomic weights for various materials, hydrogen, the starting point, being 1.

Now let us see what happens when carbon is burned to CO and CO_2. The atomic weight of carbon, from Table 15.11, is 12, and the atomic weight of oxygen with which it will combine is 16. Since the chemical symbol of carbon monoxide is CO, we know that 1 atom of carbon will join 1 atom of oxygen to form 1 *molecule* of carbon monoxide. By weight, the relationship of these atoms is of course 12 for the carbon plus 16 for the oxygen, thereby producing a molecule of CO which has a relative weight of 28. In other words, 12 lb of carbon will unite with 16 lb of oxygen to form 28 lb of carbon monoxide. Incidentally, molecular weights are developed in this way.

When the carbon burns to CO_2, the chemical symbol indicates the presence of 2 atoms of oxygen with each atom of carbon in the carbon dioxide. Consequently we have 12 parts by weight of carbon plus 16 times 2 of oxygen, making a total of 44 parts by weight of carbon dioxide.

From these relationships we readily perceive that burning carbon to

TABLE 15.11. ATOMIC WEIGHTS OF VARIOUS ELEMENTS*

Element	Symbol	Atomic weight
Aluminum	Al	26.97
Antimony	Sb	121.76
Arsenic	As	74.91
Bismuth	Bi	209.00
Cadmium	Cd	112.41
Calcium	Ca	40.08
Carbon	C	12.01
Chlorine	Cl	35.46
Chromium	Cr	52.01
Cobalt	Co	58.94
Copper	Cu	63.57
Gold	Au	197.20
Helium	He	4.003
Hydrogen	H	1.008
Iodine	I	126.92
Iridium	Ir	193.10
Iron	Fe	55.85
Lead	Pb	207.21
Magnesium	Mg	24.32
Manganese	Mn	54.93
Mercury	Hg	200.61
Molybdenum	Mo	95.95
Neon	Ne	20.183
Nickel	Ni	58.69
Nitrogen	N	14.008
Oxygen	O	16.00
Phosphorus	P	30.98
Platinum	Pt	195.23
Potassium	K	39.096
Radium	Ra	226.05
Silicon	Si	28.06
Silver	Ag	107.88
Sodium	Na	22.997
Strontium	Sr	87.83
Sulphur	S	32.06
Thorium	Th	232.12
Tin	Sn	118.70
Tungsten (Wolfram)	W	183.92
Uranium	U	238.07
Vanadium	V	50.95
Zinc	Zn	65.38
Zirconium	Zr	91.22

* *Note:* These data abstracted from the International Table of Atomic Weights for 1941.

CO_2 requires twice as much air as we need to produce CO. For example, to convert 1 lb of carbon into CO, we must join it to $^{16}/_{12} = 1\frac{1}{3}$ lb of oxygen, and the result will be $2\frac{1}{3}$ lb of CO. Now we already know that oxygen comprises only 23.15 per cent of the air by weight; therefore, in order to obtain $1\frac{1}{3}$ lb of oxygen, we must handle $1.334/0.2315 = 5.76$ lb of air.

The same procedure when applied to the process of burning a pound of carbon to CO_2 gives us $(2 \times 16)/12 = 2.667$ lb of oxygen, and $2.667/0.2315 = 11.52$ lb of air.

These are the computations that explain why the air for various combustibles must be provided in different quantities. The amount of oxygen in the end product varies; therefore, of course, the amount of air which delivers the oxygen to the reaction also varies.

The Chemistry of Hydrogen Burning to Water. Since the chemical formula for water is H_2O, we know at once that a molecule of water will be composed of 2 atoms of hydrogen and 1 atom of oxygen. In determining the air for combustion, therefore, we have these weights to work with:

Atomic weight of hydrogen........	1 × 2 atoms =	2
Atomic weight of oxygen..........	16 × 1 atom =	16
Total, molecular weight......................		18

With this ratio, 1 lb of hydrogen will require $^{16}/_{2} = 8$ lb of oxygen and $8/0.2315 = 34.56$ lb of air.

The Chemistry of Burning Sulphur to SO_2. From the symbol of sulphur dioxide, it is made up of 1 atom of sulphur and 2 atoms of oxygen.

TABLE 15.12. AIR REQUIRED FOR COMBUSTION OF COMBUSTIBLE MATERIALS

Combustible material	Lb of air per lb of combustible material	Weight of combustible, lb per cu ft
Acetylene	13.26	0.06946
Benzine	8.99	54.80000 (liquid)
Butane	15.3	0.1583
Carbon	11.52	145.0000
Carbon monoxide	3.464	0.07405
Ethane	16.07	0.08034
Ethyl alcohol	14.32	49.03000 (liquid)
Ethylene	14.32	49.03000 (liquid)
Hydrogen	34.56	0.0053
Methane	17.217	0.04245
Pentane	15.31	39.40000 (liquid)
Propane	15.58	0.11960
Sulphur	4.32	125.00000
Hydrogen sulphide	6.10	0.09616

The relative atomic weights are sulphur, 32, and oxygen, 2 × 16 = 32. For each pound of sulphur, therefore, we also need 1 lb of oxygen, and 1/0.2315 = 4.32 lb of air.

The combustion air of the liquefied petroleum gases, natural gases, and the artificial gases is uniformly calculated in this manner, starting with the chemical symbol. This is the long way around, but now that we know the way, we are justified in using Table 15.12 which, by providing us with the air needed for various combustibles, pleasantly shortens our calculations.

Products of Combustion

The gases that leave a furnace are composed primarily of the air needed for combustion, plus the various constituents of the fuel, plus whatever excess air is present.

An understanding of just what makes up these gases for any given fuel is essential for determining the efficiency of the burning process.

Suppose that we tabulate the products of combustion that result from burning oil, using the same analysis for the oil that already has served us in the calculations on combustion air. It is given in Table 15.13. This

TABLE 15.13. PRODUCTS OF COMBUSTION THAT RESULT FROM BURNING OIL

Constituent	*Per Cent*
Carbon	84.00
Hydrogen	13.00
Oxygen	1.10
Sulphur	0.40
Nitrogen	1.50

PRODUCTS FROM EACH POUND OF THIS OIL

Element	Constituent per lb of oil	Amount required to combine with element, lb		Products of combustion, lb				
		O_2	Air	CO_2	O_2	N_2	H_2O	SO_2
Carbon	0.84	2.24	9.677	3.080	.....	7.437		
Hydrogen	0.13	1.04	4.49	.....	.....	3.450	1.17	
Oxygen	0.011	.....	....	.....	0.011			
Sulphur	0.004	0.004	0.02	.....	.....	0.016	....	0.008
Nitrogen	0.015	.....	....	.....	.....	0.015		
Total	1.000	3.284	14.187	3.080	0.011	10.918	1.17	0.008
Less O_2 In Oil	.....	0.011	0.047*					
Net	.....	3.273	14.140					

* Air equivalent of the oxygen, 0.011/0.2315 = 0.047 lb of air.

tabulation shows the products from perfect combustion. Since we know perfection is beyond us for all practical operations, we can expect to find some excess air in our analysis of flue gases. For say 20 per cent excess, which is 14.140 × 0.20 or 2.828 lb, we shall find oxygen in the amount of 2.828 lb × 23.15 per cent, or 0.655 lb. The balance of the excess air, 2.173 lb, goes up the chimney as nitrogen.

Whatever our excess air happens to be, whether 1 per cent or 100 per cent, we can account for it in the products of combustion by this simple computation. Predictions, therefore, of chimney-gas conditions are quite practical if we know the analysis of the fuel.

Efficiency Determinations. The efficiency of a boiler or heater is simply the ratio of the heat we get out of it compared with the heat we put into it. Thus, if we harvest 8000 Btu from each 10,000 Btu in the fuel, we are enjoying 80 per cent efficiency; a 7000 Btu recovery means 70 per cent efficiency, and so on.

Now, let us see why we are unable to attain 100 per cent efficiency.

First, we lose heat up the stack. Second, if we are burning coal, some percentage of the carbon will escape combustion and show up in the ash. Third, there may be water in the fuel which robs our furnace of the heat needed to evaporate it and then superheat it to the temperature of the stack gases. Fourth, the moisture formed by the burning of hydrogen must be evaporated and superheated similarly. Fifth, some incompletely burned carbon may escape as carbon monoxide. Sixth, the heater or boiler radiates heat to the surrounding atmosphere. Seventh, moisture in the combustion air must be superheated, usually a negligible loss.

Some of these losses we can predict with considerable accuracy, for example, heat lost up the stack. The boiler or heater manufacturer will tell us what stack temperature to expect (if we are not satisfied to accept his figure on efficiency), and from this temperature we subtract the temperature of the combustion air. The difference represents the heat picked up by this air in the furnace and carried away outdoors. Thus, if we have a temperature difference of, say, 400 deg, each pound of the dry stack gas will lug off 400 × 0.24, or 96 Btu. Now if we are burning oil with 20 per cent excess air, our total products of combustion will weigh 17.968 lb for each pound of oil fired. However, the water vapors in these gases are deducted, since their weight in the loss line-up is treated separately. In this case we have 1.17 lb to deduct, giving us a net dry weight of 16.798 lb. The pound of fuel itself is included in this figure, since it also must be heated from the room temperature to ultimate stack temperature, where it passes off as a gas.

We now can multiply these 16.798 lb by 96 Btu per lb, and the answer is 1612.608 Btu. If the pound of fuel started into the furnace with 19,000 Btu as its heat value, then the stack loss alone represents 1612.608/19,000 = 8.48 per cent.

Now let us see what our loss is from burning hydrogen. We have 1.17 lb of water vapor, and the evaporation of this moisture as fast as it formed by the burning of the hydrogen takes 971.4 Btu per lb × 1.17 lb = 1136.54 Btu, approximately. Superheating these 1.17 lb, which have a specific heat of approximately 0.48, means 0.48 × 400 deg, or 225 Btu additional. The total loss then in Btu becomes 1136.54 + 225, or 1361.54, and dividing this total by the 19,000 Btu in the fuel indicates an efficiency loss of 7.2 per cent.

The radiation losses are quite hard to determine. Obviously, different areas on the heater or boiler present different temperatures to the fireroom. Our best plan is to obtain the recommendation of the equipment manufacturer; if we fail in this, we can estimate the heated area in square feet, assume some average temperature difference between the room and hot surfaces, and go on from there.

Heat is lost from a hot surface in two ways: (1) by radiation in the straight lines of the infrared, or "heat," rays and (2) by convection.

The heat lost by direct rays is computed by means of the Stefan-Boltzmann equation:

$$\text{Heat loss, Btu/ft}^2\text{/hr} = 0.174\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]$$

where T_1 = absolute temperature of the hot surface
T_2 = absolute temperature of the cooler surface
0.174 = a constant

To see how this works out, suppose that we estimate the heating surface at 100 sq ft, the average temperature at 200 F (the heater will be insulated), and the room temperature at 80 F. Solving the equation then becomes

$$0.174\,[(^{660}\!/_{100})^4 - (^{540}\!/_{100})^4] = 182.2\ \text{Btu/hr/ft}^2$$

Multiply by 100 sq ft, and the total radiant loss by heat rays becomes 18,220 Btu per hr.

We cannot apply this knowledge to our unit loss studies, until we learn how many pounds of fuel are burned in an hour. Since we do not have a specific job under consideration (which would tell the pounds of fuel burned per hour), suppose that we estimate 110 lb of oil as the quantity consumed. The loss per pound then by radiation would be 18,220/110, or 166 Btu.

Convection losses from flat surfaces vary with the position of the surface. If the value of 1 is assigned to the top surface of a horizontal plate in air, by comparison the loss from a vertical plate is 0.75, and the loss from the underside of a horizontal plate is only 0.50. These values of course are based on natural convection; if air is blown across the surfaces by a fan, the convection losses increase.

To calculate the loss from the heater, we need merely solve the simple equation of

$$\text{Btu per hr by convection} = uA(t_1 - t_2)$$

where u = the unit of conductance, expressed as the Btu/ft^2/hr/deg F of temperature difference
A = area of surface, sq ft
$t_1 - t_2$ = temperature difference between surface and ambient air

Some examination of the u factor is desirable, since it is affected by the position of the warm surface. Determination of the proper factor to use follows this procedure:

For top of horizontal plates............ $u = 0.4(t_1 - t_2)^{0.25}$
For vertical surfaces.................. $u = 0.3(t_1 - t_2)^{0.25}$
For bottom of horizontal plate.......... $u = 0.2(t_1 - t_2)^{0.25}$

Now, referring to the heater, or boiler, we have four vertical sides and one top side. Let us assume in regard to the 100 sq ft that 80 are vertical and 20 horizontal. There is no bottom surface exposed, as the unit is mounted against the floor.

With logarithms, we find the value of $(t_1 - t_2)^{0.25}$ to be 3.31. For the vertical areas, therefore, u will be 3.31×0.3 or 0.993; the horizontal surface will be 3.31×0.4 or 1.324.

To summarize the situation,

$80 \times 0.993 = 79.44$ Btu per deg of temperature difference
$20 \times 1.324 = 26.48$ Btu per deg of temperature difference

The total loss then from the heater surfaces by convection per degree of temperature difference is 106 Btu per hr, approximately.

Since the temperature difference is 120 deg (200 F − 80 F), the total convection loss under these conditions is 120×106, or 12,720 Btu per hr. Dividing this total by 110 lb of fuel burned per hour, the loss chargeable against each pound is 116 Btu.

The total loss from radiation and convection is $166 + 116$, or 282 Btu. This total represents an efficiency loss of $282/19{,}000 = 1.48$ per cent.

In the event the heater is situated in a drafty place, or has a fan blowing across it, the radiation loss is increased. The increase is manifest in a new u factor, computed this way:

$u = 1 + 0.22V_s$ when the velocity of the air across the surface is less than 16 fps

$u = 0.53V_s^{0.8}$ when the velocity falls between 16 and 100 fps

V_s = velocity, fps

The Emissivity Factor. Before leaving the radiation-convection loss computations, we should consider the effect of the emissivity factor on the radiation formula. In demonstrating the working of the Stefan-Boltzmann equation, in this case, the emissivity factor *e* was omitted from the demonstration, with the objective of clarifying its influence by separate examination.

The *e* factor tends to reduce the computed radiation; consequently for predictions the use of 1 as the *e* value reveals the greatest possible transmission of heat by radiation under the conditions observed.

If the surface is rough, dark, and unpolished, we can expect an *e* of between 0.90 and 1.0. This is frequently the case with our boilers and heaters; consequently the radiation values already shown in the demonstration of the Stefan-Boltzmann equation are right or nearly right. If we insert the *e* in the equation, its value would be not less than 0.90 in our problem, and the computed loss of 166 Btu multiplied by 0.90 becomes 150 Btu, approximately.

There are so many assumptions made necessarily in the estimating of radiation loss from a boiler that for all practical work the further assumption of 1 as the emissivity factor will be satisfactory, and safe.

For exact calculations in heat transfer by radiation, a table of emissivity factors for the materials involved is essential. The *e* then becomes part of the formula, which is written as

$$\text{Heat loss} = 0.174 \times e\left[\left(\frac{T_1}{100}\right)^4 = \left(\frac{T_2}{100}\right)^4\right]$$

Losses from Moisture in Air. For boiler and combustion work, this loss generally is negligible. The moisture in the air used for combustion already is in vapor form; consequently it needs only to be superheated from its entering temperature to the temperature of the stack.

The quantity of moisture in the combustion air may be read from a psychrometric chart. Thus, at 80 F and 70 per cent relative humidity, every pound of air will contain 108 grains of water vapor. With approximately 16 lb of air for each pound of oil at 20 per cent excess air operation, the total water vapor amounts to 1,728 grains, or 1,728/7,000 = 0.247 lb.

If we superheat this moisture through 400 deg, then

$$0.247 \times 400 \times 0.48 = 47.43 \text{ Btu}$$

approximately. In efficiency loss, these 47 Btu amount to less than ¼ of 1 per cent.

The Heat Balance. The ultimate step in efficiency calculations is to balance the losses against the heat in the fuel. In the case of the oil fuel we have been using as a demonstration subject, the balance is given in Table 15.14.

TABLE 15.14. BALANCE OF LOSSES AGAINST HEAT IN OIL FUEL

Heat losses in dry chimney gas	8.48%
Loss from burning hydrogen	7.20
Loss from radiation and convection	1.48
Loss from moisture in air	0.25
Unaccounted for	1.50
Total losses	18.91%
Efficiency, 100 − 18.91	81.09%

Carbon in Ash. When the fuel is coal or wood, we have another loss to consider. This loss is the result of carbon escaping combustion and being carried away in the ash.

There is no way of predicting how much this loss will be. The manufacturer of the firing equipment, whether it be a stoker, pulverized-coal system, or whatever, will guarantee the amount of this loss for his equipment, under carefully specified conditions. If we must guess at some figure for this loss, an allowance of from 2 to 3 per cent is reasonable.

The Boiler Test

The purpose of the boiler test is to obtain, or verify, the efficiency of the unit's heat transfer. All the losses that we calculated with the fuel analysis as the starting point become practical entities under the testing procedure.

The essential factor to a successful boiler test resides in our facilities for measuring everything. We must measure the amount of fuel supplied to the furnace and the weight of water fed to the boiler; we must measure the steam leaving and the air being provided for combustion; we must weigh the ash and sample it (if coal or wood is the fuel) to discover the quantity of carbon being lost via the ash pile; and we must analyze the products of combustion for carbon monoxide, carbon dioxide, and oxygen. We need the temperatures of everything, too, of air, gas, water, and fuel, for we are dealing in heat, and every Btu wherever it is must be accounted for.

Making a boiler test requires considerable apparatus and much teamwork. Particularly important is the use of an Orsat gas analyzer, rather than acceptance of simple CO_2 readings. The Orsat, if the solutions are fresh, will pick up the presence of carbon monoxide, an action beyond the capacity of the CO_2 instrument. As carbon monoxide is the result of incomplete combustion, its presence means that a combustion loss is being suffered.

The importance of CO is perhaps best appreciated if we suppose that $\frac{1}{2}$ of 1 per cent by weight appears in the flue gas. With 17 lb of flue gas per pound of fuel, the CO then amounts to 0.085 lb, with a heat value of $\frac{3}{7}$ lb of pure carbon per pound of CO. This figure of $\frac{3}{7}$ is based on the amount of carbon in a pound of CO; 12 parts are carbon, and 16, oxygen. Since a pound of carbon is worth approximately 14,600 Btu, $14{,}600 \times \frac{3}{7} \times 0.085 = 532$ Btu, approximately. In efficiency loss, this is evaluated at 2.8 per cent, a substantial figure.

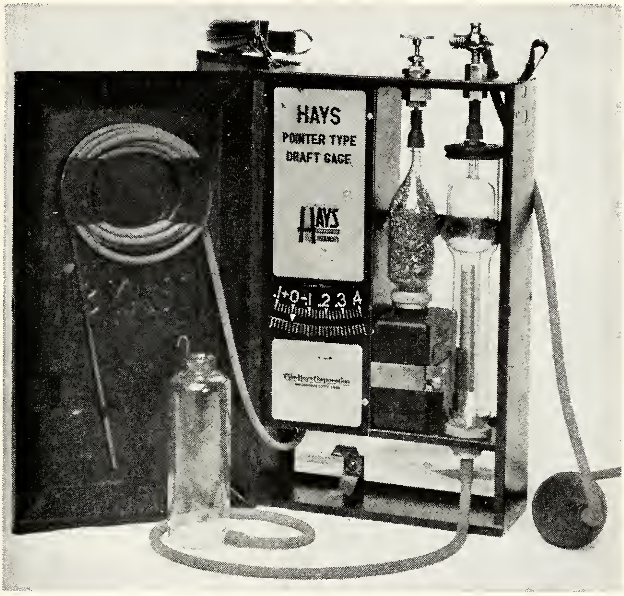


Fig. 15.3. Flue-gas analyzer for determining CO_2, draft, and stack-gas temperature. (*Courtesy of The Hays Corporation.*)

The Orsat gas analyzer is illustrated by Fig. 15.3. It is portable; it uses a solution of caustic potash for absorbing carbon dioxide, an alkaline solution of pyrogallol for oxygen, and an acid solution of cuprous chloride for the carbon monoxide.

The Significance of CO_2 Measurements. Generally, the occurrence of high CO_2 readings is considered to be indicative of good combustion. The basis for this assumption is the fact that high CO_2 occurs only when excess air is low.

Actually, we can have high CO_2 and high CO all at the same time. The CO_2 measurement, if that is all we have to go on, must be evaluated with some care for the appearance of the fire. If there is much CO going up the stack, the fire temperature falls, and instead of bright white flames yellowish hues will predominate.

A fire temperature of 3000 F represents perhaps the best practical combustion we can obtain, provided the CO_2 is high simultaneously. In many industrial plants, however, high temperatures are avoided because such heat liberation is destructive to refractories. Lower efficiency is considered by these practitioners as the lesser evil.

Low CO_2 at the boiler outlet, with good fire conditions in the furnace, indicates the presence of a leak somewhere in the setting. Air slipping in uninvited causes a loss in efficiency, because we must heat it to no useful purpose. Our duty is to plug the leak.

Apertures through the setting or casing show up when a lighted candle is passed carefully around the unit; the negative pressure within the gas passages causes the candle flame to be sucked in.

In some boilers, notably the fire-tube package-generator type, a positive pressure is commonly carried within the furnace, thus developing a pressurized form of combustion. In these designs, leaks can occur only at the ends of the boiler where the gases change direction. The escape of gases into the fireroom is readily observed; it is the result of a faulty head gasket, at these points. Corrections should be made of leaky conditions prior to running an analysis of the flue gases.

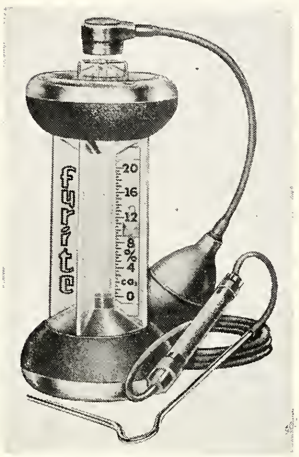


Fig. 15.4. The Fyrite CO_2 indicator. *(As manufactured by the Bacharach Industrial Instrument Company.)*

Figure 15.4 illustrates an inexpensive type of CO_2 indicator. Instruments for continuous recording of the CO_2 are installed by many boiler operators.

Smoke

What Smoke Is. Most of us are quite sure that we know smoke when we see it. Technically, however, a widely accepted definition of smoke states that "smoke is the gaseous and solid products of combustion, both visible and invisible." According to this definition, everything that combustion releases to the atmosphere is smoke.

Combustion engineers are mostly concerned with visible smoke, for this is the form on which complaints are based. Densities are evaluated by comparing the appearance of a chimney's effluent with the standard

Ringelmann charts. These charts are six in number, starting with pure white and continuing through ever-darkening gradations to No. 6, which is solid black.

Smoke inspectors will accept, generally, densities up to and including No. 3 without criticism, and No. 4 is often condoned for short periods. Heavier smoke constitutes a definite nuisance.

Figure 15.5 is a reproduction of the Ringelmann charts.

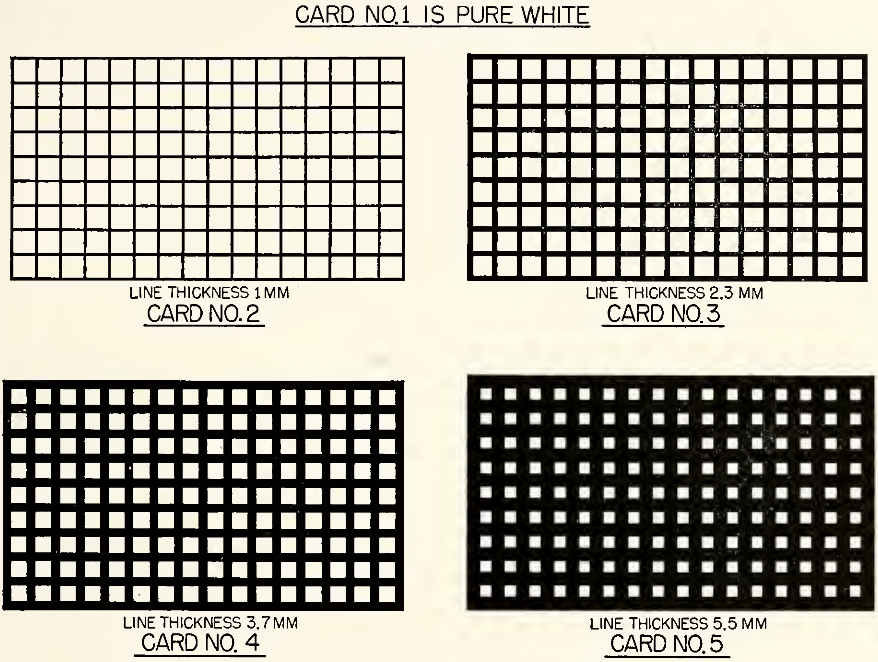


Fig. 15.5 Ringelmann charts showing comparative density of smoke.

Causes of Smoke. The cause of smoke is closely connected with the character of the fuel, the design of the furnace, and the skill of the individual furnace operator.

Suppose that we consider bituminous coal first. This fuel, predominant in the steam generating field, enters the furnace possessing large percentages of volatile matter. Before ignition temperature is reached by the fresh coal, these volatiles separate and go up the chimney. In a well-designed furnace with ample volume, many of the volatiles are consumed before reaching the first bank of tubes; if the design is unfavorable, heavy smoke is the result.

Burning Volatiles. Furnace designs attempt in several ways to combat smoke formation and to burn the volatiles usefully. Figure 6.2 (Chap. 6) illustrates the so-called "ignition arch of refractory," intended to ignite the gases as they leave the coal on entering the furnace. This arch is customarily used with flat grate stokers, and overfeed stokers. Underfeed stokers do not employ the idea, since the green coal enters the furnace from underneath the fire, and the gases must pass through the fire to escape.

Another successful plan introduces air or steam into the furnace from nozzles usually mounted in the bridge-wall areas but sometimes in the front wall above the stoker. These wall nozzles may be found serving all types of stokers, even those already furnished with an ignition arch.

The purpose of the jet is to provide violent turbulence and intimate mixing of the gases with combustion air. As a result, most of the volatiles are consumed before they can leave the furnace.

Air jets have established satisfactory operating results for many previously condemned offenders. Locomotives in particular have employed the jets to eliminate smoke nuisances.

The amount of the air needed and pressure of injection vary with each installation. However, for a starting point, we might expect to introduce 25 per cent of the combustion air through the nozzles, and at a pressure of 2 or 3 in. W.G. The pressure, however, is a function of the air-tube size and the distance of penetration by the jet into the furnace. Large furnaces naturally will require greater pressures if the turbulence is to affect the entire grate area; for these applications, air pressures up to 20 in. are not unusual.

The ideal manner of controlling the jet operation is by means of a photoelectric cell. When the cell "sees" smoke over the fire, in the breeching or at the stack outlet, depending on our choice of location, it turns on the air to the jets. When the smoke ceases, the jets are shut down. It is all automatic and the vagaries of human control are avoided.

Figure 15.6 shows a simple installation of jets in a bridge wall.

Complete data on the use of jets may be obtained from *Technical Report* VIII, "Application of Overfire Jets to Prevent Smoke from Stationary Plants," issued by Bituminous Coal Research, Inc., Pittsburgh, Pa.

Steam instead of air has been used with some success. However, nozzle erosion seems to make necessary frequent replacements, and in many plants diversion of the steam quantities required cannot be suffered.

Effect of Furnace Design. Furnaces are no larger than needed, and consequently forcing boilers to meet extra-heavy loads almost inevitably creates smoke. The cure of course is a larger boiler or the installation of jets. Another alternative is to replace a coal firing system with oil since the removal of stokers and grates produces an increase in the clear volume of the furnace.

All of a furnace volume is not usable for combustion, as we can see by tracing the gas movements from the fuel to the boiler heating surface. The objective is to design a furnace which will employ as much of the volume as possible. With oil fuels particularly, a simple relocating of the burner has been known to stop smoke; more of the furnace volume is being put to work. Even the projection of the burner into the furnace can cause trouble if the distance between the burner nozzle and the opposite wall is critical.

The fact is, if we make an ill match between the firing equipment and the furnace volume or arrangement, we can expect to produce a smoky chimney.

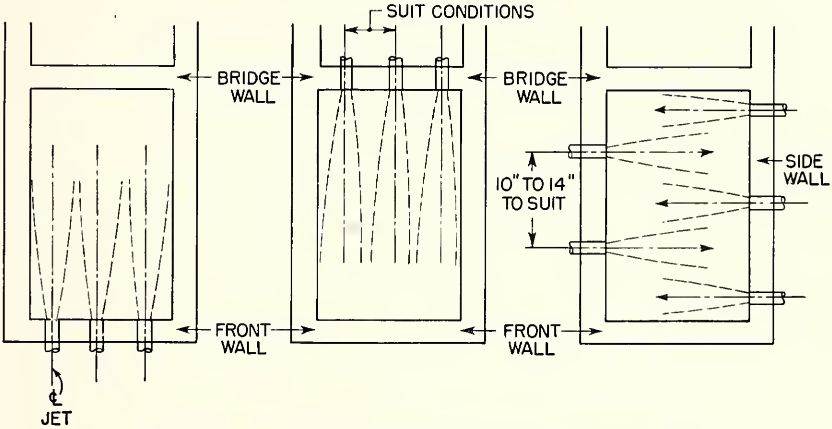


Fig. 15.6 Steam jets in boiler walls to eliminate smoke.

The Human Element and Smoke. Firemen who attempt to increase stoker capacities by shoveling coal into the furnace by hand, tossing the fuel on top of the fire, only increase the number and intensities of the neighborhood complaints. Such practices add tarry soot and cinders to the stack gases, and tend to precipitate legal action against the plant.

An installation that tempts operating personnel into such acts is seriously in need of study. For some reason or another, the combustion capacities are being overloaded; alternately, the operators are grossly incapable, and should be instructed otherwise or replaced.

When oil is being burned, the presence of smoke indicates insufficient air, dirty burners, too-viscous oil, or faulty atomization. If the smoke is dense white, it signifies a very large percentage of excess air.

All these conditions are within the ability of a competent operator to correct. Although white smoke not often is reported as a nuisance, the efficiency loss actually may be greater than if the effluent were jet black.

Incidentally, the efficiency lost due to carbon in heavy black smoke is estimated by combustion experts to be approximately 1 per cent.

When we summarize the smoke situation, we observe that its treatment involves a study of the fuel, the furnace, and the practices of operation. Many times we can improve all three.

Questions

1. Describe to a friend the burnable elements in coal and oil.

2. Why does the burning of carbon to carbon monoxide signify a loss of efficiency?

3. How much air is needed to burn a pound of carbon? a pound of hydrogen? a pound of sulphur? a pound of methane?

4. Obtain an analysis of the fuel you are burning in your home. How much air should you provide for perfect combustion?

5. From the analysis of your own fuel, calculate the products of combustion.

6. Why do wood and bagasse require large percentages of excess air?

7. If you were making up a heat balance for your own fuel, what losses of efficiency would you be required to consider and evaluate?

8. What is the purpose of the boiler test?

9. What is the significance of white smoke?

10. If a chimney persistently is emitting heavy black smoke, where would you first look for the cause?

11. What would you do if the cause of smoke was found to be one of the following?
    - *a.* Inadequate furnace volume for the quantity of fuel being burned.
    - *b.* An oil burner poorly mounted with relation to the shape of the furnace.
    - *c.* Dirty or poor-grade fuel.
    - *d.* Faulty operating practices.
12. Plan a boiler test to check the efficiency of a newly installed boiler.
    - *a.* Itemize the instruments needed.
    - *b.* Determine the number of men needed, and tabulate the duties of each.

CHAPTER 16

CHIMNEYS, STACKS, AND FLUES

Principles of Design

The Names They Go By. While the names of "chimney" and "stack" frequently have been and are used interchangeably, there seems to be a growing tendency to distinguish them. Current practice refers to domestic and low-pressure boilers as being served by chimneys, while industry almost invariably speaks of its towering smoke pipes, both brick and steel, as stacks.

Regardless of the term we use, the basic principles of design are identical. Construction of course introduces variations in the physical structure, depending on the materials and type of chimney or stack that we select. Both design and construction will be discussed in detail.

The Two Basic Factors in Natural Draft Design. The first factor we must evaluate is the theoretical draft, or gross draft, available. This gross draft results, theoretically, from the difference in density between air at the bottom of the chimney and air at the top. Since air becomes less dense as (1) it is heated and (2) it is moved upward from the surface of the earth, our gross draft depends partly on the difference in temperature between the flue gas and the outside air and partly on the height of the chimney above the fire. There is also the effect of altitude above sea level to consider, if we are located on a mountain or high plateau, but this effect will be discussed later, since it is really a modifier of the theoretical draft and not a basic principle.

The second basic factor we must evaluate is the loss by friction through the boiler, breeching, and the chimney itself. By subtracting this total friction loss from the gross or theoretical draft, we arrive at the "net available draft." If our net available draft is insufficient to maintain combustion as energetically and as efficiently as we desire, then we must increase the height of our chimney, provide it with greater cross-sectional area, and perhaps do both.

These are the two factors that govern the height and diameter of our design.

Calculating Natural Draft. Suppose that we consider the draft generated by a chimney 100 ft high. The items we must coordinate to determine the theoretical draft are the following:

H = height of the chimney; in this example 100 ft
P = atmospheric pressure; in this example 14.7 psig
T = absolute ambient temperature
T_1 = absolute temperature of hot gases entering chimney

For the example, we will assume that it is June outdoors and the temperature is 80 F; the temperature of the chimney gas we will take as 450 F.

The relationship of these various items is shown in the draft formula as follows:

$$\text{Draft} = 0.52HP\left[\frac{1}{T} - \frac{1}{T_1}\right]$$

Substituting the assumed figures for temperature, etc., the solution becomes

$$D = 0.52 \times 100 \times 14.7[1/540 - 1/910]$$
$$D = 0.574 \text{ in. W.G.}$$

This calculated draft is our theoretical, or gross, draft for the chimney in the example.

The effect of altitude above sea level is felt in this formula by the P item of atmospheric pressure. At 1,000 ft elevation for example, the atmospheric pressure is not 14.7 psig, but 14.2. The theoretical draft is reduced accordingly.

In our example, by solving the equation on the basis of the 14.2 psig value for atmospheric pressure, the draft falls to 0.555 in. W.G. This is not too serious, and in general we can use the sea-level pressure up to 1,000 ft elevation with safety, since we normally allow some sensible factor of safety, and do not attempt to design right up to the last thousandth of an inch.

Table 4.7 (Chap. 4) proffers information on the atmospheric pressures at elevations up to 20,000 ft. We should use it when calculating draft for locations above 1,000 ft.

Calculating Friction. The total friction loss which the draft must overcome, if we are to get the products of combustion out of the furnace, includes the following items:

1. Friction of air moving through grates and fuel bed.
2. Friction of products of combustion moving through the boiler.
3. Friction of the hot gases moving through the breeching to the chimney.
4. Friction of the hot gases moving upward through the chimney itself.

Item 1 is for furnaces burning solid fuel on a grate. Since the friction varies as the result of the thickness of the bed, the presence of clinker, the fusing nature of the fuel, and similar conditions that change constantly, exact predictions are impractical, and some figure must be assumed for the loss. The following figures show the trend of this friction value:

Boiler operating at designed rating..........	0.30 in. W.G.
Boiler operating at 150% rating.............	0.60 in. W.G.
Boiler operating at 200% rating.............	1.00 in. W.G.

If the boiler is burning anthracite coal, we might start with 0.4 in. at boiler rating and end up with 1.2 in. W.G.

Item 2 normally will be provided for us by the builder of the boiler. Friction in the boiler is affected by the baffling; consequently accurate evaluations cannot be made without a study of gas passes, temperatures, and changes of direction that are essentially functions of the boiler design, and have been computed and balanced by the designer.

In the event that no figures are available from the builder, we can arrive at preliminary estimates suitable for striking a tentative chimney design by assuming the following:

Boiler operating at designed rating..........	0.25 in. W.G.
Boiler operating at 150% of rating...........	0.40 in. W.G.
Boiler operating at 200% of rating...........	0.70 in. W.G.

Items 3 and 4 are calculated from the same friction formula. The formula is

$$\Delta D = \frac{fW^2CH}{A^3}$$

where ΔD = friction loss, in. W.G.

f = a friction factor that varies with the temperature of the gas and the material surface over which the gas is flowing

W = weight of gas passing up the chimney, lb per sec

C = perimeter of stack, ft (inside)

H = height of stack, or length of breeching, ft

A = average cross-sectional area of gas passage, sq ft

Since the friction factor is a variable, it poses some problems of accuracy in its choice, but again we must realize that a factor of safety is desirable in all empirical studies. Consequently, we can select a friction factor from the following figures, using a straight-line relationship for intermediate temperatures:

Stack surface	Gas at 350 F	Gas at 600 F
Steel..............	0.0011	0.0015
Brick..............	0.0015	0.0020

One other factor enters into the computation, and that is the matter of turns in the breeching. Every time the hot gases turn a corner, they introduce another item of friction. For a steel breeching, this loss amounts to approximately 0.05 in. W.G. in each right angle, or 90-deg turn. If the breeching is built of brick or concrete, the increased roughage of the surface imposes on us the need for doubling this allowance.

Sometimes in domestic installations some offset of the flue is encountered, particularly where fireplaces are concerned, and these offsets should be valued at 0.05 in. per offset, since the tile flue is approximately equal to steel, or smoother, from the standpoint of friction.

As an example of how the friction is computed, we assume that there is a brick chimney or stack 100 ft high serving a boiler through a steel breeching 20 ft long and with two 90-deg turns.

We must calculate the weight of the products of combustion, W, in the friction formula, and in addition must determine the perimeter of the inside surfaces of chimney and breeching.

Chapter 15 discusses in detail the calculating of the products of combustion for various fuels; for the moment we will assume that we are burning coal with 20 lb of hot gas for every pound of coal fired. If we are burning 1,000 lb of coal per hour, then we must handle 20,000 lb of hot gas per hour, or 5.56 lb per sec, approximately. These 5.56 lb become the W of the formula.

Reaching an estimate for the perimeter requires us to give some thought to gas velocities. For natural draft, something between 10 and 20 fps is found in good practice. Suppose that we therefore use the mean of 15 fps.

Referring to Table 2.7 (Chap. 2) we observe that air at 450 F, which is our assumed gas temperature, occupies 22.9 cu ft per lb, which means that our 5.56 lb of flue gas presents a volume for handling of 127.324 cu ft. At a velocity of 15 fps, the stack area becomes 127.324/15, or 8.5 sq ft. By solving the simple arithmetical formula of area $= \pi R^2$, the diameter of the chimney, inside, is found to be 39 in., and the perimeter ($\pi \times$ diameter) is 10.2 ft.

Now we can set down the figures we know in the friction formula as follows:

$$\Delta D = \frac{0.0017 \times 30.91 \times 10.2 \times 100}{614.125}$$

$$\Delta D = 0.087 \text{ in. W.G.}$$

So much for the chimney; now let us see what happens in the breeching. The formula remains basically the same, but we must change our friction factor to compensate for the use of a steel surface, and the H in the formula becomes the length of the breeching, in this case 20 ft. Consequently, the revised figures are

$$\Delta D = \frac{0.00126 \times 30.91 \times 10.2 \times 20}{614.125}$$

$$\begin{aligned} \Delta D &= 0.013 \text{ in. W.G.} \\ &+ 0.100 \text{ in. W.G. allowed for two 90-deg turns} \\ &\overline{0.113} \text{ in. W.G.} = \text{total loss in breeching} \end{aligned}$$

Please note that in calculating the friction in the breeching we have assumed the breeching to be round with a perimeter equal to that of the chimney. This was done in the example for convenience and to avoid the introduction of additional calculations which would follow if we had to determine the perimeter of a rectangular breeching or one of other shape.

While the perimeter is important in determining the actual area of the surfaces over which the gas must pass, our prime concern in designing a breeching is to avoid any reduction in cross-sectional area as compared to the chimney. Thus in the example worked out, so long as we provide 8.5 sq ft of cross-sectional area in the breeching, many minor sins in the shape of the breeching can be forgiven. Some designers attempt to provide an area in the breeching somewhat greater than in the chimney, perhaps up to 20 per cent more, working on the theory that the gas in the breeching is hotter than in the stack and therefore requires more space for its movements.

Undoubtedly a generously designed breeching is an asset, but in view of the space limitations currently encountered by many boiler installations even the minimum is not always achieved except by a determined and ingenious struggle. We seldom can hope for more.

The Summary of Friction Losses. Where a theoretical draft and the friction that this draft will be asked to overcome have been figured, the balance may be set down as in Table 16.1.

TABLE 16.1. SUMMARY OF FRICTION LOSSES

Theoretical draft		0.574 in. W.G.
Friction of air through fuel bed	0.300	
Friction loss through boiler	0.250	
Friction loss through breeching	0.113	
Friction loss in chimney	0.087	
Total	0.750 in. W.G.	

Obviously, we shall need more draft. We can obtain it, of course, by adding a forced or induced draft fan, but since we are engaged at the moment strictly with natural draft, we must extend the height of the chimney.

Some discussion of the boiler rating concept is in order at this time. It is luck if most hand-fired boilers produce the rating their builders have made them capable of, considering the care they receive. For this reason, the loss through the boiler has been assumed to occur at the designed rating in the balance we have just struck. For industrial installations, it is prudent to base the calculations on at least 150 per cent of the designed rating, and 200 per cent would not be too far out of line, since industrial boilers notoriously operate well above the arbitrary capacities of their design.

One exception to this overrating should be observed: In the case of the fire-tube steam generators, conventionally referred to as "package boilers," there is seldom more than 10 per cent extra steam to be squeezed from the manufacturer's rating. Steam releasing space in these units is limited; consequently any substantial acceleration in the formation of steam bubbles inevitably hurls water out of the boiler and into the steam piping.

Industrial boilers, also, unless they have been operating these many years, are quite sure to be equipped with draft fans. These fans greatly reduce the height of stack necessary; in fact with the package generators, it is nothing more than a vent pipe, and is not charged with any responsibility for producing draft.

Draft for oil- or gas-fired boilers is less than we need for coal, since these fluid fuel users pose no fuel bed for the air to penetrate. The same reasoning holds true for spreader stokers and pulverized-coal burning, where most of the combustion takes place with the coal in suspension.

There remains one precaution to be noted before we leave the computing of natural draft. It is this: We must never expect to realize more than 80 per cent of the calculated theoretical draft.

There are so many transient forces that adversely affect the draft of a chimney, such as wind movement, the barometer changes, even birds' nests clinging to the inside surfaces, that simple prudence requires us to discount our expectations.

An ineffective chimney is an abomination to all concerned.

The Rules of Practice

The Underwriters' Rules. The National Board of Fire Underwriters in attempting to safeguard structures from chimney fires has set down some exact rules that we should observe. The complete Underwriters' story is one of great detail, and should be obtained in its entirety by all of us engaged in chimney design. Some of the important points to be considered by the designer are the following:

1. Chimneys should extend at least 3 ft beyond the roof through which they pass, and in any event should be carried not less than 2 ft higher than any roof ridge which exists within 10 ft of the chimney.
2. Chimneys should be wholly supported on masonry or self-supporting fireproof construction.
3. Unlined chimneys should have walls not less than 8 in. thick, if built of brick, sawed and dressed stone, concrete, or burned clay; but rough stone should be 12 in. thick.
4. Chimneys in residences and small similar buildings, having linings of approved flue tile, may be built with brick walls approximately 4 in. thick.
5. Flue linings shall be at least $\frac{5}{8}$ in. thick for the small flues, with the thickness increasing as the size of the flue increases. Table 16.2 lists the standard sizes of flues.
6. Flue linings should start at least 8 in. below the breeching connection, and should extend 4 in. above the top of the chimney. Figure 16.1 illustrates flue positions in a chimney.
7. Linings should be manufactured of refractory fire clay good for temperatures of at least 2000 F. Alternatively, cast-iron liners may be used, if the quality, form, and construction satisfy the requirements of the National Board of Fire Underwriters Code.
8. A masonry cap should be used around the top of the chimney to seal the brick construction around the projecting flue lining and to prevent the leaking in of water around the liner.

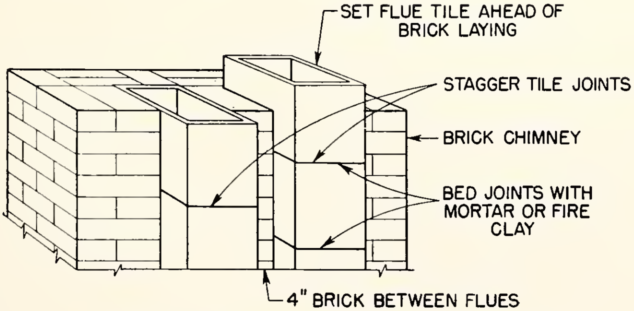

SET FLUE TILE AHEAD OF BRICK LAYING
STAGGER TILE JOINTS
BRICK CHIMNEY
BED JOINTS WITH MORTAR OR FIRE CLAY
4" BRICK BETWEEN FLUES

Fig. 16.1 Arrangement of flues in a chimney.

The Rules from Experience. Through the centuries of chimney building, the following rules of satisfactory practice have been formulated more or less painfully, and should be observed by those of us who want to avoid some personal discomfort and criticism:

1. Support the chimney by a spread footing, being careful to locate the said footing beneath the frost line.

2. If we feel the necessity of surmounting the chimney with a rain cover, either in the form of an arch or with a flat slab supported some inches above the end of the flue lining, then the total area of the openings under the cover and through which the smoke will escape *must* add up to four times the net area of the flue. Anything less will affect adversely the natural draft of the chimney.

3. The heating plant must have its own flue. Attempting to share the heating flue with other processes of combustion will bring dissatisfaction to all concerned.

4. The mortar used for laying up the flue liners, or the brick courses if the chimney is unlined, should be waterproofed. Soot frequently contains considerable sulphur, which, when wet by rain, becomes a sulphuric acid in effect and promptly attacks the mortar. This results in leaky flues and perhaps falling bricks.

5. Chimneys should be filled in from the foundation right up to a point not over 12 in. below the connection of the breeching. Hollow bases, by filling with cold, dense air, adversely affect the draft.

Table 16.2. Recommended Sizes for Flues

Capacity of connected steam radiation, sq ft	Inside min dimensions, in.		Min effective flue area, sq in.
	Round	Rectangular	
590	10	8½ × 13	70
690	10	13 × 13	79
900	12	13 × 13	100
1,100	12	13 × 17½	113
1,700	15	13 × 17½	150
1,940	15	17½ × 17½	177
2,130	15	17½ × 17½	183
2,480	18	20 × 20	234
3,150	18	20 × 24	234
4,300	20	24 × 24	314
5,000	20	24 × 24	346

The Breeching and the Underwriters. The metal pipe that connects the boiler or furnace to the chimney can be a seriously hostile piece of equipment when improperly designed or installed. Here are some rules set up by the National Board of Fire Underwriters to avoid fires starting from hot breechings:

1. Provide at least 24 in. clearance to the nearest exposed woodwork.
2. Provide 18 in. of clearance to a surface formed of wood lath and plaster; at least 9 in. if the lath is incombustible.

3. Allow 1-in. clearance for masonry and concrete.
4. The metal thickness should be at least No. 24 gauge.
5. A sheet-metal or asbestos panel should be affixed in most installations to the wooden rafters or joists above the heater and breeching. This panel is intended to prevent ignition of the joists from excessive heat or sudden puffs and flashes, which occasionally occur as the result of unfavorable fuel and draft combinations.
6. While cement-asbestos and porcelain enamel breechings are sometimes used, the approval of the National Board of Fire Underwriters should be obtained in each case.

Cleaning the Chimneys and Breechings. A simple way to clean a chimney is to lower a brick on a piece of wire, swinging it back and forth. Care must be taken, however, not to break the flue tile by overvigorous impacts from the brick.

A better way is to hire chimney cleaning experts. These individuals will do a rapid and effective job with vacuum apparatus.

The use of chemicals for cleaning chimneys should be undertaken only with the greatest caution. Some of these chemical agents if applied to soot at high temperatures may provoke an explosion or intensive and dangerous combustion. Other soot cleaning materials offered to the trade have been found to be valueless.

Before using any chemical cleaner, the approval of the National Board of Fire Underwriters should be obtained.

Breeching cleaning is an easier job. Frequently the entire breeching, especially in a domestic installation, is easily removed, brushed out, and replaced.

In larger plants, where the breeching is a sizable and heavy contraption, and usually insulated with magnesia and asbestos, the designer normally provides a cleanout door, conveniently located in the side or bottom. This door should be large enough to admit a hoe, a long-handled wire brush, and a man's head, although not all of these simultaneously, and the last is desirable but not vital. The head is inserted for the purpose of seeing where the dirt lies and if the cleaning job is being accomplished efficiently.

Lightweight Chimneys

What They Are. In the ever-continuing effort to reduce construction costs, especially for the owner of a small home, we now have recourse to a chimney of comparatively recent design that saves money by avoiding the need for a foundation. This chimney is so light in weight that it can be suspended from the framework of the building. It is approved by the National Board of Fire Underwriters.

The design of this lightweight chimney is illustrated in Fig. 16.2. It is built of metal cylinders (in one design aluminum), one within the other, and the space between is filled with insulation, additionally cooled by an air bath.

Variations of this arrangement are being exercised by different manufacturers, but the end result is similar; all are light in weight and are free of masonry.

The trend toward small, one-story basementless houses, which are obliged to yield a portion of their already scanty floor space to a heating plant, is giving impetus to the use of these chimneys. By suspending them above the heater, there results a saving not only in investment costs, but in floor space as well.

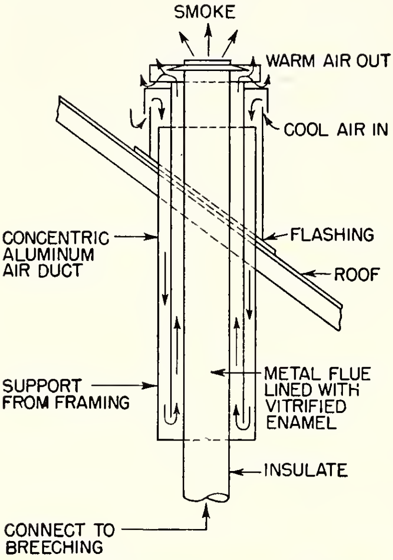


Fig. 16.2. Thulman lightweight chimney formed of aluminum with vitrified fire tube. (*As built by Lennox Furnace Company, Inc.*)

Commercial and Industrial Chimneys

The Design of Independent Brick Chimneys. Many large heating plants and industrial boilers are served by one or more brick chimneys, often standing at some distance from the building, and posing major problems of design and construction.

The ultimate details of such chimneys should be delegated to experts in their construction, but the design engineer in the heating field requires a sufficient knowledge of their principles and characteristics to specify what is needed and wanted.

The best design for a tall brick chimney is really a double design; that is, it comprises an outside brick envelope and a separate inside tube or core. The inside tube, preferably of firebrick throughout, is not anchored to the outside envelope, as the expansion stresses would be quite certain in a tall chimney to pull apart the outside brick if a tie were used.

The foundations for a brick chimney are of first importance, as any tilting, uneven settling, or cracking of the foundation would imperil the integrity of the entire structure. A rule of thumb suggests a safe limit of 2 tons per sq ft if the soil is gravelly, but compact. If the chimney can be located to rest on a foundation of rock, so much the better.

The thickness of the brick varies from bottom to top in any chimney of height, and for a chimney, say, 100 ft high, it may start with 16-in. walls for the envelope at the bottom and grade down to 8 in. at the top. The thickness of the inner tube is designed with similar gradations. As a general rule, we might expect the taper in wall thickness to average from $\frac{1}{16}$ to $\frac{1}{4}$ in. per ft of height, depending on the ideas of the designer and the wind stresses the chimney is likely to encounter.

For a quick estimate of the proportions of a chimney, custom suggests the use of a base diameter which is $\frac{1}{10}$ of the height. This is the outside diameter, while the inside diameter depends, as we have noted earlier in this chapter, on the factors of theoretical draft and friction.

The wind pressure on any chimney may be calculated from the formula:

$$\text{Pressure, psf} = 0.004V^2$$

As V in the formula is velocity of the wind in miles per hour, a 10-mile wind develops a pressure on a flat normal surface of 0.4 psf; a 100-mph hurricane imposes 40.00 psf.

In calculating the pressure on a round chimney, we first determine the projected area. This is accomplished by the simple procedure of multiplying the outside diameter by the height. Now we apply the wind pressure in pounds per square foot against this projected area, and the result is, theoretically, the total wind pressure on the chimney.

Actually, the shape of the chimney surface affects the wind pressure quite substantially, since little ridges and depressions cause eddies that tend to relieve the pressure in their respective neighborhoods. In consequence, the total *effective* pressure seldom or never achieves the full theoretical value, and should be estimated at the percentages shown in Table 16.3, according to Kent:

TABLE 16.3. ESTIMATED TOTAL EFFECTIVE PRESSURE

For round chimneys	67%
For hexagonal perimeters	80%
For octagonal perimeters	71%
For normal flat surfaces	100%

Wind velocity of 100 mph is a sound basis for chimney design, although in some cities the design velocity, either higher or lower than this, is set by law. For any projected chimney, the local authorities should be consulted before actual design begins.

Foundations for brick chimneys are not affected by the wind pressure to any great extent, since the chimney is not integral with the foundation. In consequence, foundations for brick chimneys are designed to support the weight, to spread it reasonably over the soil so that settling does not occur unevenly, and beyond this duty they have none.

The Design of Self-supporting Steel Chimneys. Since steel stacks are cheaper than brick chimneys to build, we see many of them. Their conventional design employs steel plate in courses not over 72 in. high, and the courses are formed to permit the telescoping of each into the bottom of the course next above it. The joints then are welded both inside and outside, although the inner welds frequently perform no more than a sealing service.

The bottom of a self-supporting stack is belled out to increase its stability, the flare usually extending upward to about one-fourth of the stack's total height, and starting with a diameter in the proportion of 4 to 3 or 1.5 to 1, compared with the cylindrical diameter. Older stacks may show a shorter flare, perhaps only one-seventh of the total height, since this relationship was common practice a quarter century ago.

Plate thickness also is heavier in the flare, usually 1⁄16 in. thicker than is employed in the next adjacent section, and never should be less than 5⁄16 in., with ¼ in. as a practical minimum in the subsequent sections.

TABLE 16.4. ANCHOR BOLTS FOR STEEL STACKS

Stack		Anchor bolts	
Diameter, ft	Height, ft	No.	Thickness, in.
4	100	16	1⅜
5	125	14	1⅞
6	125	24	1⅜
6	150	18	1⅞
7	150	18	1⅞
7	175	20	2⅛
8	175	20	2⅛
8	200	20	2½
9	200	24	2⅛
10	200	26	2⅛

Self-supporting stacks are anchored securely to their foundations with anchor bolts in a ring at the bottom of the flare. The number of anchor bolts, their thicknesses and lengths, may be calculated, or we may take the easier road of accepting common practice. Thicker bolts are spaced on greater centers than thin bolts, a 1⅞-in. bolt, for example, will be used approximately on 12- or 13-in. centers, a 1⅜-in. bolt on 9½-in. centers. Some illustrative data on anchor bolts, as employed by one builder of stacks, are given in Table 16.4.

Data on calculating the size and number of anchor bolts, incorporating tensile strength of material and various stresses, may be obtained from any standard handbook.

TABLE 16.5. APPROXIMATE WEIGHTS OF SELF-SUPPORTING STEEL STACKS AS COMPUTED BY KENT*

Height, ft	Diam., ft	Height of flare, ft	Approx. weight, lb
100	4	30	20,000
125	5	30	32,800
125	6	40	35,690
150	7	40	55,740
175	8	40	80,950
200	9	55	108,900
200	10	50	119,000
225	12	60	165,000
250	12	65	206,800

* Abstracted by permission from the "Buffalo Handbook," 5th ed., Buffalo Tank Corporation.

Foundations for Self-supporting Chimneys. The anchor bolts which hold steel chimneys firmly to the foundations transmit the overturning effect of the wind to the foundations, and consequently we must design our concrete to resist successfully such untoward results.

There are several approaches to the problem of determining the diameter of the foundation, the depth of the concrete being reached by the simple procedure of multiplying the diameter by 0.40, since that is the standard relationship of the two dimensions.

There is an old Kent formula for finding foundation diameters that works swiftly and effectively. Here it is:

$$d = 1.07 \sqrt[4]{DH^2}$$

where d = the diameter of the foundation, ft
D = the diameter of the cylindrical stack, ft
H = the height of the stack, ft

For example, suppose that we have a stack 100 ft high and 4 ft in diameter. Substituting these values in the formula, we have

$$d = 1.07 \sqrt[4]{4 \times 10{,}000}$$
$$d = 15 \text{ ft}$$

Multiplying 15 by 0.40 = 6 ft as the depth of the concrete.

Now let us check the effectiveness of this block of concrete in its duty of preventing the wind from blowing over our stack. The overturning moment of the wind at the top of the foundation and bottom of the stack is

$$M = \frac{pdh^2}{2}$$

where M = the overturning moment, in-lb
p = wind pressure, psf × 0.67
d = diameter of stack, ft, cylindrical section
h = height of stack, or distance between top of stack and any point lower down at which we wish to compute the bending moment

If we are designing our stack against a wind of 100 mph velocity, then the effective pressure per square foot is $100 \times 100 \times 0.004 \times 0.67 = 27$ psf. Inserting this pressure with our other figures in the bending-moment formula, we have

$$M = \frac{27 \times 4 \times 10{,}000}{2}$$

$$M = 540{,}000 \text{ in-lb}$$

The next step is to evaluate what we know, and have, against conventional practice which calls for a factor of safety of at least 2.5. Here is the formula we must solve:

$$d = \frac{F_s M}{W_s + W_b + W_c}$$

where d = diameter of foundation, ft
F_s = factor of safety
M = bending moment, in-lb
W_s = weight of steel stack, lb (see Table 16.5. Note, the extra weight of the flare usually is neglected)
W_b = weight of brick lining, if any, lb
W_c = weight of concrete foundation, lb

Since we now have all the figures, except that of the factor of safety, which we want, we can write our data into the formula and solve it. Thus

$$15 = \frac{F_s \times 540{,}000}{20{,}000 + 0 + 159{,}300}$$

$$15 = F_s \times 3.0 +$$

$$F_s = 5 \qquad \text{approximately}$$

This factor of safety appears to be ample for the job; in fact one prominent builder of stacks cuts down his depth of concrete from two-fifth the diameter of the foundation to one-third the diameter.

Table 16.6 is faster and on the whole more certain than the method of computing given, since every stack builder relies largely on his own experience. The foundations given in this table have been found by long practice to be quite safe for wind pressures up to 25 psf.

Another formula for computing the diameter of a foundation is

$$\text{Diameter, ft} = \frac{8M}{W_s + W_b + W_c}$$

This formula employs a very high factor of safety and results in a much larger foundation than is shown in Table 16.6.

Table 16.6. Diameters of Circular Masonry Foundations for Self-supporting Steel Stacks*
(Height of foundation = ⅖ diam. Pressure = 25 psf)

Height of stack, ft	Diameter of stack, ft							
	4	5	6	7	8	9	10	12
70	14.3	15.1	15.8	16.4	17.0	17.5	18.0	18.8
80	15.3	16.1	16.9	17.6	18.2	18.7	19.2	20.1
90	16.2	17.1	17.9	18.6	19.3	19.8	20.4	21.3
100	17.1	18.0	18.9	19.6	20.3	20.9	21.5	22.5
110	17.9	18.9	19.8	20.6	21.3	21.9	22.5	23.6
120	18.7	19.8	20.7	21.5	22.2	22.9	23.5	24.6
150	20.9	22.1	23.1	24.1	24.9	25.6	26.3	27.5
180	....	....	....	26.4	27.2	28.1	28.8	30.1
200	....	....	....	....	28.7	29.6	30.4	31.8

* Abstracted by permission from the "Buffalo Handbook," 5th ed., p. 153, Buffalo Tank Corporation.

Steel Stacks with Guys. The guyed stack produces a saving in investment costs by (1) not requiring a heavy foundation and (2) using thinner and less metal than a self-supporting stack.

Some designers prefer to thin out the metal as the stack nears its maximum height, but actually corrosion is more industrious in the top sections, owing to the greater incidence of moisture, and a uniform thickness generally pays for itself.

Except perhaps in very short stacks of 25 ft or less, the metal thickness should be of at least a quarter inch. The stacks are fabricated similarly to the self-supporting stacks, using sections or courses not over 72 in. high, and with each course telescoping into the bottom of the one above it.

Guys may be attached to the stack by either of two methods: (1) by the use of an eyebolt for each guy, backed up by a reinforcing plate inside the stack, and (2) by attaching the guys to a circumferential band. The former method is preferred, since it is less subject to deterioration from corrosion.

The number of sets of guys for any given stack depends upon the height. For a stack 50 ft high, one set is enough, attached two-thirds of the way to the top. For 75 ft, we need two sets, the additional set being tied to the stack's middle. For 100 ft or more, three sets are customary, each set being located to meet approximately one-third of the total strain.

The guys themselves may be formed of wire, galvanized wire rope, or rods. Nothing thinner than $\frac{3}{8}$ in. should be considered safe.

Table 16.7 provides information on the construction of these stacks as conventionally followed.

Table 16.7. Stack Guys and Plate Thicknesses*

Diam., in.	Height of stack, ft									
	25		50		75		100		125	
	No. of sets	Thick., in.	No. of sets	Thick., in.	No. of sets	Thick., in.	No. of sets	Thick., in.	No. of sets	Thick., in.
18	1	$\frac{3}{16}$	1	$\frac{1}{4}$						
24	1	$\frac{3}{16}$	1	$\frac{1}{4}$						
36	. .	. . .	1	$\frac{1}{4}$	2	$\frac{1}{4}$	3	$\frac{5}{16}$	3	$\frac{5}{16}$
42	. .	. . .	1	$\frac{1}{4}$	2	$\frac{1}{4}$	3	$\frac{5}{16}$	3	$\frac{5}{16}$
48	. .	. . .	. .	. .	2	$\frac{5}{16}$	3	$\frac{5}{16}$	3	$\frac{5}{16}$
60	. .	. . .	. .	. .	2	$\frac{5}{16}$	3	$\frac{5}{16}$	3	$\frac{5}{16}$
72	. .	. . .	. .	. .	. .	. . .	3	$\frac{5}{16}$	3	$\frac{3}{8}$

* *Note:* Three guys normally constitute one set, but this number may be increased if conditions warrant.

Painting of Metal Stacks. The inside surfaces of metal stacks should be painted with a good grade of heat-resisting corrosion-resisting paint. Several coats are desirable. Most fuels, and particularly coal, are likely to contain a measure of sulphur, and this sulphur when combined with oxygen and moisture forms compounds that attack industriously the ferric surface of a stack.

Outside surfaces must withstand the weather, plus the heat from the gases inside the stack, and the paint should be chosen to meet these conditions. Aluminum paint produces a nice appearing stack, and in some industrial atmospheres resists corrosion better than any other finish.

The existing ambient conditions should be understood thoroughly before selecting a paint, since some materials which perform satisfactorily in one location may fail completely in another.

Stacks of Radial Brick. These chimneys are formed of a special brick, shaped to satisfy the radius of the chimney. The bricks are perforated,

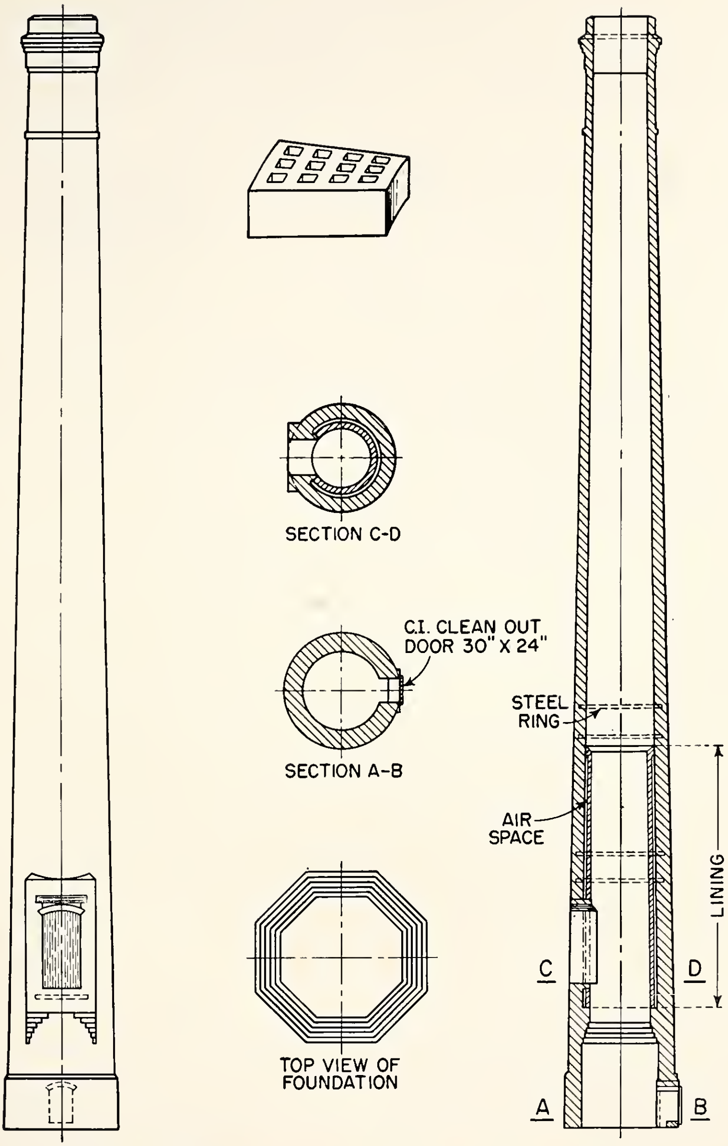


Fig. 16.3 Heinicke perforated radial brick chimney.

the holes being about 1 in. square. The bricks are about 4½ in. thick, with radial lengths from 4 to 10½ in., approximately. Observe Fig. 16.3.

Chimneys of Reinforced Concrete. Concrete chimneys are particularly desirable in earthquake country, where they resist earth vibrations more successfully than brick. The chimney is built by pouring the concrete into adjustable steel forms, which are moved up, section by section, as the structure rises.

Concrete chimneys taper at a rate of 3¾ in. on the diameter for each 10 ft of height, with a top wall thickness of 5 or 6 in.

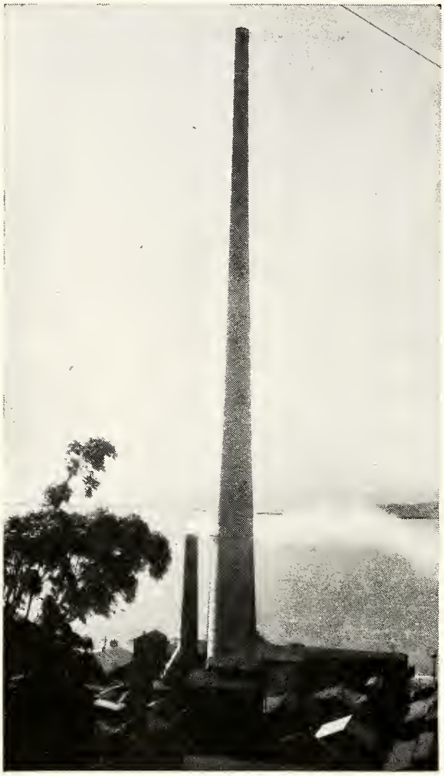


Fig. 16.4 Reinforced concrete chimney as erected at Selby, Calif. *(Courtesy of Custodis Construction Company.)*

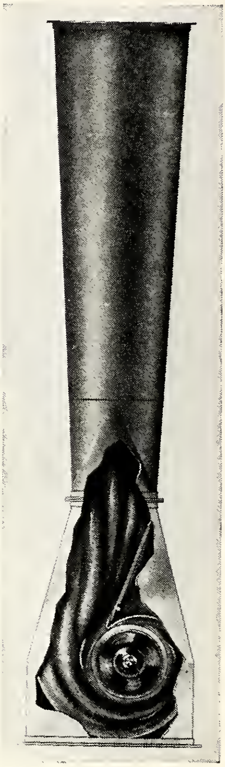


Fig. 16.5 Position of induced draft fan in venturi stack. *(Courtesy of Prat-Daniel Corporation.*

Because of the comparatively elaborate equipment needed by a reinforced concrete chimney, a radial brick chimney may be cheaper up to, say, 150 ft in height.

Figure 16.4 illustrates the extreme heights to which concrete chimneys may go, this one being 600 ft high, 13 ft inside diameter at the top, 43 ft 11½ in. outside diameter at the bottom. Top wall is 7 in. thick, bottom, 25 in. Total weight above foundation is 5,200 tons.

Ladders should be built of ¾-in. round steel, preferably 16 in. wide, and with a safety cage up to the gas-sampling platforms, if these sampling stations are provided.

Reinforcing is both vertical and horizontal.

When designing a concrete chimney, we should consult experts, and in all cases the American Concrete Institute, for specifications.

The Venturi or Evasé Type of Stack. Figure 16.5 illustrates a form of stack designed for service with an induced draft fan exclusively.

The principle of operation is observed to be the familiar one of the ejector, in which a fast-moving jet of steam or air is introduced at the narrow end of an expanding tube, and thereby creates a suction through the tube.

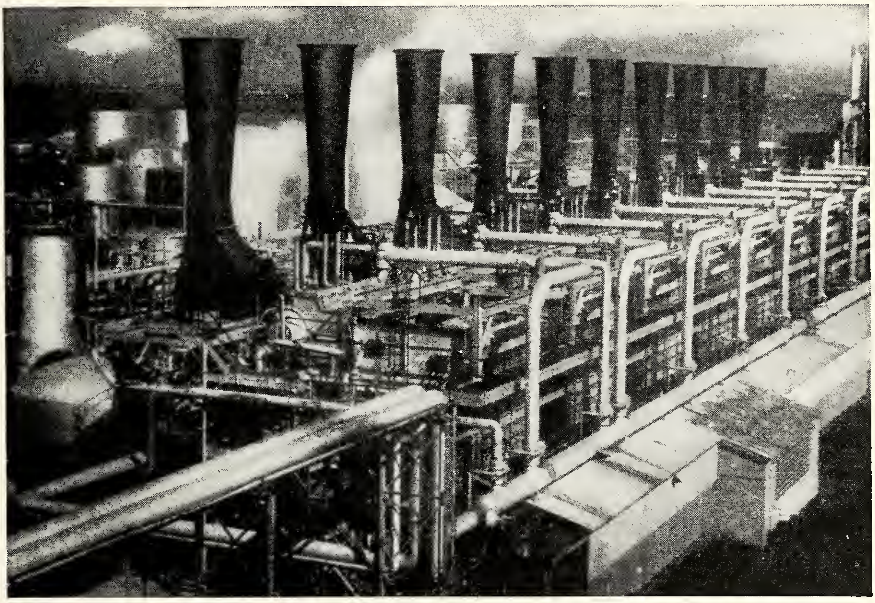


Fig. 16.6. Group of venturi stacks in a chemical plant. *(Courtesy of Prat-Daniel Corporation.)*

In the venturi stack, the high velocity jet may be formed with a portion of the hot gases abstracted from the flue by the induced draft fan, or the fan may build the jet from fresh air. The latter method is easier on the fan from the standpoint of temperature, gas density, and fan erosion, but the injection of cold air drops the gas temperature in the stack, causing a reduction in the draft produced.

The venturi arrangement materially reduces the horsepower required to drive the induced draft fan, a fact that justifies its serious consideration by designers. Perhaps the major disadvantage is observed in the shortness of the stack, measured beyond the throat, for brevity in handling, when smoke and cinders are concerned, is not always the soul of perfection in the eyes of our neighbors.

The venturi stack of course is reserved for service with large commercial or industrial installations. It embodies just too much equipment for residential acceptance. Figure 16.6 shows an industrial installation.

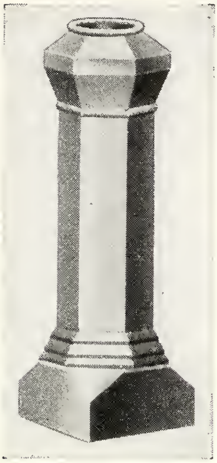


Fig. 16.7. Clay-tile chimney pot. Available in various shapes and sizes. (*Courtesy of Robinson Clay Product Company.*)

Chimney Terminals

Chimney Pots and Such. The addition of clay chimney pots to chimney design seems to have aimed at these targets: (1) the reduction in the amount of rain that can enter a large chimney; (2) improvement in draft by somewhat extending the height of the chimney, and simultaneously making the entry of downdrafts more difficult; (3) the substitution of a smart appearance for what was all too often merely a sooty and black-ringed hole.

We can assume that the rain reduction and the appearance betterment objectives have been achieved, but the affect of a chimney pot on chimney draft is not always beneficial. Chimney pots tend to restrict the smoke outlet, and if draft conditions in a chimney are critical to start with, a handsome pot at the top of the flue may affect quite unhandsomely the heater or boiler at the bottom of the flue. When this happens, amputation of the chimney pot is the simplest and cheapest treatment.

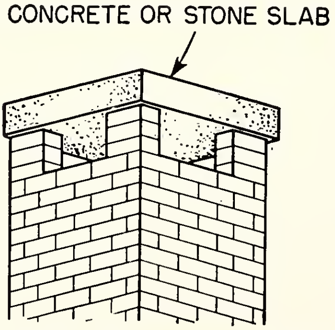


Fig. 16.8. Flat type of flue cover.

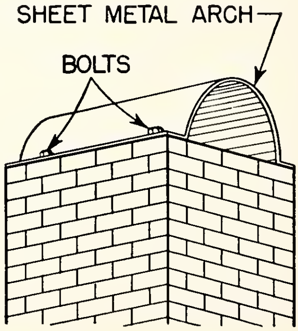


Fig. 16.9. Arch type of flue cover.

Figure 16.7 illustrates a type of chimney pot that is pleasing to architects and homeowners alike. Before the heating system designer accepts this or any other form of chimney dressing, he should check carefully the natural draft available, and weigh it against the friction losses introduced by the proposed beautification.

Figure 16.8 shows the flatcap type of flue covering, and Fig. 16.9 the arched design, both of which tend to interfere with chimney draft *unless* the ultimate openings for exit of the smoke add up in total area to *four times* the area of the flue beneath. This 4 to 1 ratio has been overlooked by many chimney builders, and the error is a prolific cause of draft complaints from the operators of heaters and fireplaces.

Sheet-metal chimney extensions, often seen if the chimney construction has been stopped below the ridge line of the roof, or if the house is overshadowed by trees, are intended to stop downdrafts. They also stop observers with their appalling ugliness.

Rather than afflict a building with such unsightliness, draft corrections should be achieved by an extension of the chimney, or if that is impractical by the installation in the basement or heater room of a draft fan.

Vents for Gas-fired Furnaces

Characteristics of Gas Combustion. The air needed for burning gaseous fuel, whether carburreted water gas, natural gas, or liquefied petroleum gas, enters the burner from the ambient atmosphere and at atmospheric pressure. No fans or chimney drafts are needed to promote combustion; a gas burner will function in a closed room as long as oxygen in sufficient quantity endures.

Chapter 15 on combustion describes the products of combustion which result from the burning of gas, and we observe that a substantial weight of water vapor is prominent among them. This vapor is what causes the feeling of moist heat that gradually develops in a room around an open gas flame when the room is closed. Carbon dioxide, nitrogen, and perhaps a trace of carbon monoxide if combustion is inefficient also pollute the atmosphere of a room under nonventilated conditions. Because these products of combustion are invisible and nonodorous, their presence is sometimes overlooked; but they are dangerous and any gas-fired heating system should be designed to deliver them firmly and promptly to the great outdoors.

The Draft Hood. Since a gas flame is easily extinguished by a puff of air, the gas burners in a heating boiler or furnace must be protected against the sharp downdrafts that sometimes dip into a flue and are not stopped until they reach the fire itself.

The draft hood, illustrated in Chap. 4, is designed to divert these descending winds before they can reach the gas flame and blow it out. The draft hood simply directs the wrong way gusts harmlessly into the room.

Draft hoods are not used with large industrial boilers, since a positive or at least uninterrupted removal of the flue gas is essential. In these cases the employment of an induced draft fan is common, or, if natural draft is depended upon, the draft hood is replaced by some form of automatic balanced draft regulator.

A type of burner known as the "power burner" is also employed in industrial installations and it actually develops a degree of positive pressure in the furnace which is sufficient to combat downdrafts effectively. This burner is described in detail in Chap. 6, which is concerned with firing equipment.

The draft hood is required on all domestic boilers that carry the approval of the American Gas Association Laboratories.

Gas Flues and the National Board of Fire Underwriters. The National Board of Fire Underwriters tells us that each of the following gas appliances, except kitchen ranges, shall have an effective flue outlet for the products of combustion:

1. When rated in excess of 50,000 Btu per hr.
2. If controlled automatically and releasing more than 5000 Btu per hr.
3. Appliances in the same room which have an aggregate demand at normal rating equal to or exceeding 30 Btu/hr/ft^3 of room content.

The flues themselves shall meet the following specifications:

1. Lined stacks or metal stacks must be provided for incinerators and for all appliances which produce flue-gas temperatures in excess of 550 F at the outlet of the vent connection.
2. The flues must be formed of incombustible, noncorroding material of sufficient thickness, cross-sectional area, and heat insulating quality to avoid temperatures in excess of 160 F on adjacent combustible material.
3. If flues are formed of soil pipe or steam or water pipe, with acceptable joints, and are located inside partitions or walls, or extend through a wall to the outside, they should be encased in an incombustible jacket with a 1-in. air space extending the entire length, and the jacket or duct must be open at both top and bottom.

The National Board of Fire Underwriters issues a pamphlet "Recommended Good Practice Requirements," which we will be wise to consult from time to time.

Sizing the Flues. A simple way of sizing a chimney or flue for a gas-fired boiler is to allow 1 sq in. of cross-sectional area for each 6500 Btu of *input*. Thus, if our heater input is 200,000 Btu per hr, we shall need a flue of 200,000/6,500 = 31 sq in., approximately. The diameter of a circle 31 sq in. in area is about $6\frac{3}{8}$ in.; consequently we will use a 7-in. flue. Figure 16.10 shows a height, load, diameter relationship.

Sometimes we have need to vent two or more appliances to the same chimney or stack. In these cases we determine the size of the flue for each appliance in square inches of cross-sectional area and then add these areas together, and the answer is the required area of the stack.

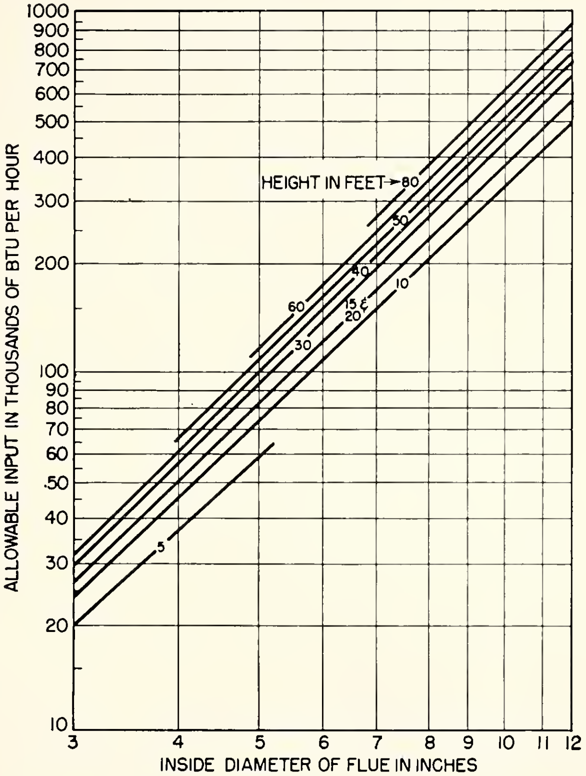


Fig. 16.10. Height, load, and flue diameter relationships.

Some Rules of Good Practice. The American Gas Association, after considerable research, has published a set of rules that wisdom directs us to observe when designing our gas flues. Some of these rules are given in gist herewith:

1. Construct the flue or stack of a good, heat-insulating material. We can choose from clay tile, asbestos, cement pipe, glazed sewer pipe, or metal pipe. Other materials have proved satisfactory in special locations.

2. Eliminate horizontal runs if at all possible. Consistently pitch upward at a rate of 1 in. every 2 ft for a minimum, and approach the

vertical as nearly as conditions allow. *Never grade downward.* In calculating resistance, remember that a 90-deg elbow injects the resistance of 1 ft of straight pipe for each 1 in. of the elbow's diameter. Thus a 5-in. elbow is the frictional equivalent of 5 ft of straight pipe.

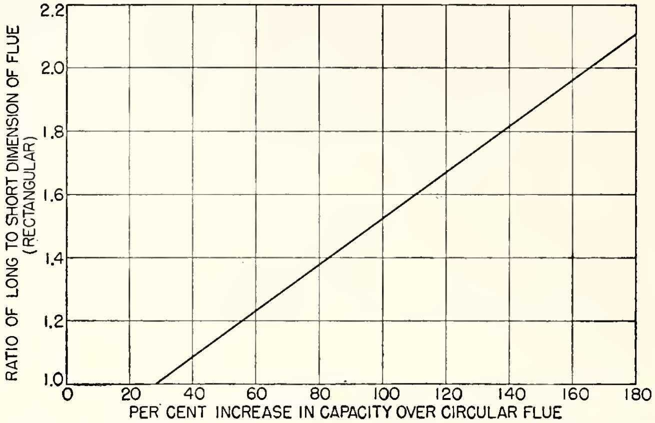


Fig. 16.11. Capacities of rectangular gas vents. *(Reproduced by permisson of the American Gas Association.)*

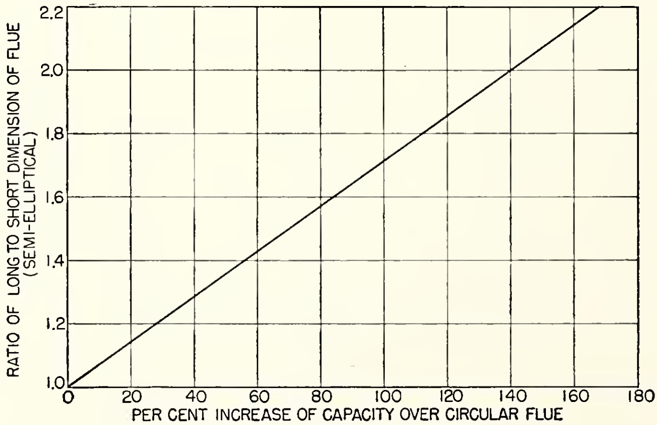


Fig. 16.12. Capacities of oval gas vents. *(Reproduced by permission of the American Gas Association.)*

3. In converging flues, use a Y or 45-deg junction, never pattern on the T.

4. Locate the draft hood where the appliance manufacturer stipulates. If the appliance bears the approval label of the American Gas Association Laboratories, the location has been tested and approved.

5. As in the case of all chimneys, carry the vent or stack at least 2 ft

higher than any roof, wall, or other object that is present within 20 ft of the vent, and threatens to become a potential blanket.

Vent Shapes. Most vents are of circular cross section. Table 16.8 records their standard sizes. However, it is possible to obtain vents that are oval, or basically rectangular in cross section, with smoothly rounded corners. The rectangular and oval vents have a somewhat greater venting capacity, as may be seen from an inspection of Figs 16.11 and 16.12.

Table 16.8. Round Flue Capacities, Btu Input, for Domestic Gas-fired Appliances with Draft Hoods*

Inside diam. of flue, in.	Height of vent flue, ft		
	30	50	100
3	20,000	27,000	30,000
4	44,000	51,000	60,000
5	75,000	87,000	105,000
6	118,000	130,000	155,000
7	178,000	190,000	230,000
8	220,000	250,000	305,000
9	280,000	330,000	400,000
10	350,000	420,000	500,000

* These capacities have been compiled from data obtained from the American Gas Association and have reference to appliances with draft hoods. For commercial installations with tall, high capacity stacks, refer to "Gas Engineers' Handbook," published by McGraw-Hill Book Company, Inc., 1934. Temperatures contemplated in this table are 150 F for the combustion gases and 60 F for the outside air. In addition, excess air is considered to be 100 per cent with an additional 100 per cent dilution occurring in the draft hood.

Chimney Troubles

A brief review of chimney troubles and their elimination will be helpful not only in dissolving difficulties with existing structures, but as a guide when new projects are in the making. These troubles manifest themselves by the appearance of smoke at the wrong places and at the wrong times, by a failure of the chimney to develop adequate draft, or perhaps by a fire. Here is an analysis:

Poor Draft. This may be cured by the following means:

1. Carry the chimney at least 2 ft above the roof ridge, or an equal distance above any other sizable object within 20 ft.
2. Remodel or remove the chimney cap, if any.
3. Provide unrestricted exit area at the top of the chimney.
4. Clear the chimney of internal obstructions, broken tile, brick, birds'

nests, etc. If floor joists are found projecting into the chimney, remove them before they burn themselves off and possibly burn the house down.

5. If more than one flue is in the chimney, check for possible cross leakage between them. One course of brick between flues is strongly recommended for all new construction.
6. Smoke pipe, or breeching, either may extend into the chimney or fit so loosely that air leakage occurs around it. Close leaks with cement. Shorten the breeching as necessary.
7. There may be air leakage through the cleanout door in the bottom of the chimney. Stop it.
8. If the breeching enters the chimney high up, a long hollow void below the breeching creates a cushion of cold air that seriously detracts from the draft. Fill the void.

Smoke. This may be cured by the following actions:

1. Verify the size of the chimney against the weight of the products of combustion it must handle. A chimney that is too small reflects on the designer and can be corrected only by introducing positive draft, or by rebuilding.
2. Clean the chimney.

Downdrafts. These may be cured by the following expedients:

1. Increase the height of the chimney.
2. Install either the flat or arch type of chimney cap, but the smoke openings *must* equal four times the cross-sectional area of the flue.
3. Install some type of positive draft apparatus, such as a fan.
4. Mount a patented exhauster on the chimney (Fig. 18.3, Chap. 18).

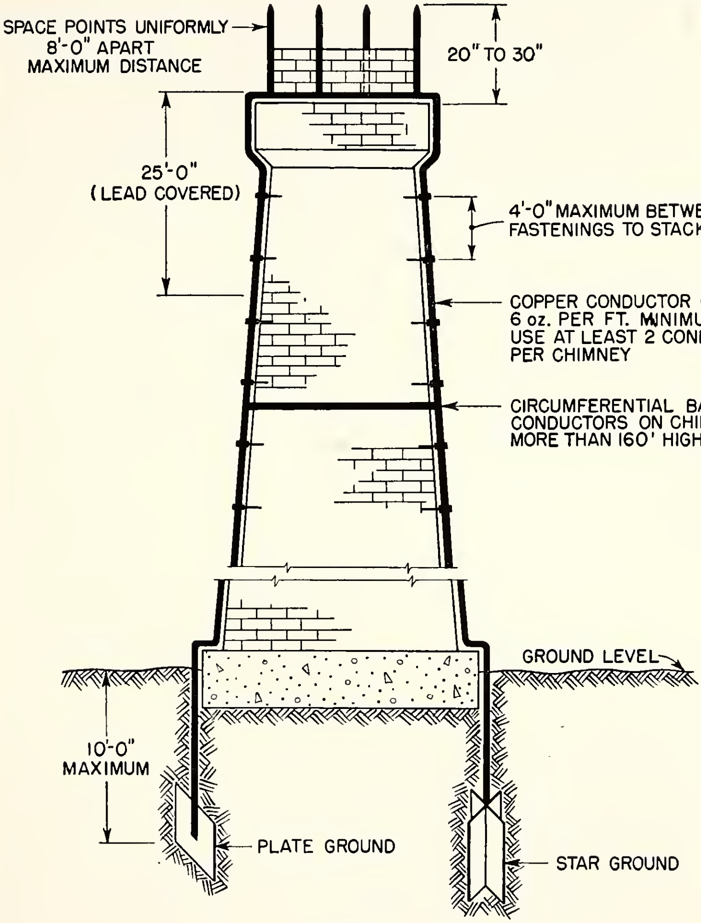
Chimneys and Lightning

Chimneys are hit with lightning rather more often than other parts of a structure, generally because they rear themselves so predominantly into the blue. For this reason, if we protect our chimneys against the bolt, we are quite likely to protect surrounding buildings as well.

Chimneys require a little special treatment when it comes to mounting the lightning rods, because the hot gases are corrosive and the terminals in the corrosive areas must be encased in some protective material such as lead.

Large chimneys, say more than 24 in. in diameter, ordinarily require two air terminal rods with a conductor that encircles the top. The down cable should be fastened into the brick, and not into the mortar, if the chimney is built of brick. The conductor should penetrate the earth by at least 10 ft, and keep as far away from water piping or other underground metal structures as may be possible.

For the chimney illustrated in Fig. 16.13, six points are spaced at equal intervals around the top with three down-leading cables that terminate in copper-covered grounding rods, $\frac{3}{4}$ in. in diameter, which are driven 10 ft into the ground. The down-leads are interconnected every 50 ft by circumferential cables. The points and top 35 ft of the down-leading cables are sheathed in $\frac{1}{16}$-in. lead as a protection against corrosion.


SPACE POINTS UNIFORMLY 8'-0" APART MAXIMUM DISTANCE
20" TO 30"
25'-0" (LEAD COVERED)
4'-0" MAXIMUM BETWEEN FASTENINGS TO STACK
COPPER CONDUCTOR CABLE 6 oz. PER FT. MINIMUM WEIGHT USE AT LEAST 2 CONDUCTORS PER CHIMNEY
CIRCUMFERENTIAL BAND TIES CONDUCTORS ON CHIMNEYS MORE THAN 160' HIGH
GROUND LEVEL
10'-0" MAXIMUM
PLATE GROUND
STAR GROUND

NOTE:

1- BOND METAL BREECHING TO CONDUCTOR SYSTEM
2- CONSULT EXPERIENCED CONTRACTOR FOR DETAILS
3- EITHER TYPE OF GROUND PLATE MAY BE USED

Fig. 16.13. Installation of lightning rod on chimney.

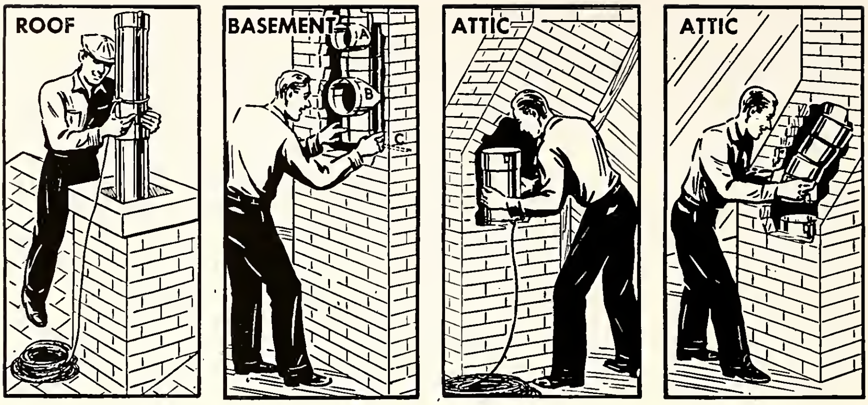


Fig. 16.14 The vitroliner flue showing installation procedure (*Courtesy of Condensation Engineering Corp.*)

The cost of protecting an ordinary residential chimney is likely to be in the neighborhood of $100. For industrial stacks, the investment is considerably greater, but when compared with the expense of building a new chimney, the cost of lightning protection is usually justified.

For steel stacks, adequate grounding is the only protection needed.

New Lining for Old Chimneys

The current popular practice of modernizing old colonial homes frequently raises a question as to the safety of the ancient, usually unlined, stone or brick chimney. The wear and tear of a century or more may have introduced a crack here and there, through which smoke, gases, and perhaps a spark or two might find a passage to the living areas of the house.

The introduction of vitrified tile from the top of the chimney to create a liner for the full length of the flue is quite practical in these cases. At least one manufacturer specializes in flue linings of this sort, and has developed material that may be used with complete satisfaction, despite the existence in the flue of offsets and angles. Figures 16.14 and 16.15 illustrate the way it is done.

The cost of installing a single flue of this kind, for a chimney, say, 50 ft high, may run from $60 to $100,

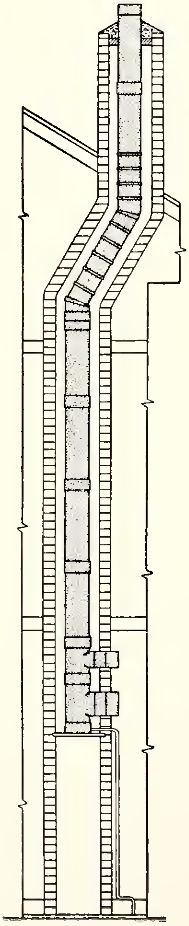


Fig. 16.15. The vitroliner flue in place.

depending on the size of the liner required and the stubbornness of the difficulties encountered.

Questions

1. Name the two basic principles of chimney design.

2. Calculate the theoretical natural draft and the friction loss in the chimney of the building in which you work or live. Does this calculation explain any draft troubles that may exist?

3. How many of the Underwriters' rules do you remember? How many of the rules of practice?

4. Enumerate the Underwriters' rules for safe breechings.

5. How would you clean the chimney in your own house? Have you ever looked into it, to ascertain its condition?

6. What are the advantages of a brick chimney?

7. If you were building a new plant, would you choose brick, concrete, or steel chimneys? Why?

8. How does a self-supporting steel stack differ from a guyed stack?

9. Compute the size of the foundation needed for a steel stack 200 ft high and 10 ft in diameter. Will it withstand the overturning moment of a 100 mph wind?

10. What is the wind pressure for governing stack design, set by ordinance in your particular city?

11. Where would you use a draft hood for a gas-fired furnace?

12. Write up the rules of good practice for gas-fired installations.

13. What would you do to cure poor draft in a domestic chimney?

14. Would you design lightning protection for a high chimney? Why? Can you justify the expense for the chimney you have in mind?

CHAPTER 17

HEAT INSULATION AND REFRACTORIES

Commercial Insulations

The never-ending need for heat-arresting materials has developed for us an almost incredibly wide choice of blankets, preformed shapes, reflectors, concretes, and plasters from which to satisfy our needs. These needs are in themselves variable; consequently we may state in general that any given insulation chosen for some particular application may fail to satisfy the demands of another. Our first step then is to determine precisely what we want from an insulation, and in many cases the simple arresting of heat transmission will be discovered a lesser requirement.

These needs may be clarified if we examine and evaluate the marks of a good insulating material, and weigh them against our particular problem.

The Marks of a Good Insulation. 1. We conventionally desire a material that shows a low factor of heat conductivity, expressed as the *k* factor by the trade. This characteristic obviously was the first ever required of an insulating material, since it is a direct measure of a substance's ability to arrest the passage of heat.

2. Equally important is the insulation's ability to maintain its high heat resistance down the years. We expect to apply insulation to a given job just once, and if the *k* factor grows substantially with time, that material is not for us. In short, we want insulation that does not deteriorate.

3. In some installations, noncombustibility is a must. This means that some of the fire-resistant or fire-retardant insulations, despite their attractive *k* factors, cannot be used.

4. Moisture seriously lowers the heat resistance of many air-cell materials, or destroys it completely. This moisture does not originate in water impact only, it may accumulate gradually, worming its way into the insulation from a consistently humid atmosphere. In tropical areas, therefore, we must be sure that our insulation is water sealed, or choose a different type.

5. Density, as a weight to be handled, is of no consequence as a rule, except aboard ships or airplanes, where all nonpayloads are avoided as far

as possible. Actually, conductivity appears to increase with the density, indicating that insulating virtue resides in a multiplicity of voids.

6. Ease of installation, which aids in holding down the investment cost, is always desirable.

Suppose we now examine some of the familiar insulating materials from the viewpoint of desirable qualities.

Rock Wool. Most of our high temperature insulations are formed under this name. Rock wool really is a slag, said to have originated in the Hawaiian volcano Kilauea. Presently it is being manufactured from both metal and mineral slags, is of various shades of gray to black, and is recommended for contact with hot surfaces right up to 1800 F.

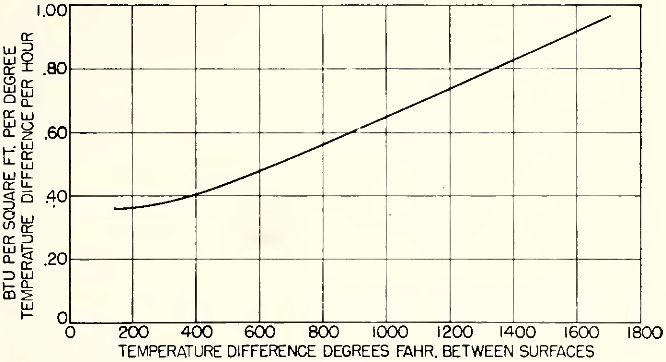


Fig. 17.1. Characteristic curve of Hi-Temp insulation conductivity.

The conductivity of rock wool, as with all insulating materials, increases as the temperature rises. Figure 17.1 shows a typical trend of the k-factor values over the full operating range. Density is in the neighborhood of 20 lb per cu ft.

Recommended thicknesses for the application of rock wool are as shown in Table 17.1. Rock wool is noncombustible, and its response to moisture is negligible. It does not deteriorate to any notable extent with time, so long as the temperature for which it is designed is not exceeded.

TABLE 17.1. RECOMMENDED THICKNESSES OF APPLICATION OF ROCK WOOL

Temperature of Hot Surface, F	*Thickness, In.*
Up to 300	1½
300– 450	2
450– 600	3
600– 800	3½
800–1000	4
1000–1200	4½
1200–1400	5
1400–1800	6

Asbestos Sponge Felt. The top limit of temperature for this material is 700 F. It is formed by felting, weaving, or compressing asbestos fibers and tiny particles of sponge.

The conductivity factor may be represented by a straight line over the full range of temperatures starting with 0.45 at 200 F and ending with 0.64 at 700 F.

Asbestos sponge felt is resistant to shock, vibration, and moisture. It is a common choice for underground pipe lines and for various railroad installations. It does not deteriorate with age.

Asbestos sponge felt is recommended for application to indoor pipes and ducts in accordance with the data in Table 17.2.

TABLE 17.2. RECOMMENDATIONS OF ASBESTOS SPONGE FELT FOR APPLICATION TO INDOOR PIPES AND DUCTS

Temperature of hot surface, F	Thickness of insulation, in.		
	Large pipes and flat surfaces	Pipe diameter, in.	
		2–4	under 2
Up to 266	1	1	1
267–387	2	1½	1
388–499	2½	2	1½
500–599	3	2½	2
600–700	3½	3	2

For outdoor service, each of these thicknesses should be increased by ½ in. Density is approximately 24 lb per cu ft.

85% Magnesia. This long accepted insulation is composed of 85% magnesium carbonate and 15% binder, usually asbestos fibers.

The top temperature for 85% magnesia service is 600 F, and its k factor, like that of asbestos sponge felt, runs in an approximately straight line from 0.47 at 200 F to about 0.70 at 600 F. Magnesia may be soaked in water, dried, and reused without any diminution of effectiveness. It is not hygroscopic, and consequently offers no problem in tropical climates.

The thickness of 85% magnesia generally is referred to by the trade in terms of "standard" and "double standard thick." The standard thickness is a nominal 1 in. that becomes actually ⅞ in. for pipes 1½ in. in diameter and smaller, 1$\frac{1}{32}$ in. for pipes 2 to 3½ in. diameter, 1⅛ in. for pipes 4 to 6 in.; 1¼ in. for pipes 7 to 10 in. and finally 1½ in. for all larger pipes. The double standard is approximately double these actual thicknesses.

The 85% magnesia usually is purchased in preformed blocks, averaging 36 in. long. These blocks are available also in various curved and tapered shapes to fit valves and fittings. Canvas jackets and brass lacquered joint bands are furnished with the blocks.

Recommended thicknesses for 85% magnesia application are given in Table 17.3. Where more than one thickness is needed, the insulation should be applied in the broken joint arrangement, that is, the junctions of the blocks should not coincide either longitudinally or circumferentially. The joints themselves should be sealed with 85% magnesia cement.

TABLE 17.3. RECOMMENDED THICKNESSES FOR APPLICATION OF 85 PER CENT MAGNESIA IN INCHES

Temperature of hot surface, F	Flat surfaces	Pipe diameter, in.		
		Up to 2	2–4	Over 4
Up to 267	Standard	Standard	Standard	Standard
267–338	1½	Standard	Standard	1½
339–388	2	Standard	1½	2
389–500	2½	1½	2	Double standard
501–600	3	2	Double standard	3

Lacing wire used for holding the blocks in place should be either of copper or annealed steel wire, 16 gauge in weight, and spaced approximately 12 in. on circumferential centers. The final jacket frequently is pasted on for low-temperature heating work, but should be sewed for high-pressure steam lines, using 8-oz duck.

When the insulation must serve under outdoor conditions, general practice calls for an additional layer of standard thickness, and the application, over the canvas jacket, of three-ply asphalt roofing with the joints lapped at least 3 in. for watertightness. The asphalt roofing is wired on, using 16-gauge copper wire on 8-in. centers. Each lap should be pasted down with an asphalt cement prior to the wiring.

Asbestos Air-cell Insulation. Here is the material we use probably more than any other, for the average residential heating system, where the temperatures involved are less than 300 F.

A section of asbestos air cell is shown in Fig. 17.2. It is formed of corrugated asbestos paper, usually in either three- or four-ply arrangements. The over-all thickness of the three-ply insulation is ¾ in., of the four-ply, 1 in.

The sections are finished on the outside with either a canvas jacket, cemented to the asbestos, or a specially treated white paper which is proof against water, oil, dust, and grease.

The k factor may be taken at 0.65 for most calculations, this being its rating at 200 F. For hot surface temperatures up to 165 F, the three-ply thickness generally is sufficient, the four-ply material being recommended for all higher temperatures.

Air-cell insulation presents a very smart finish, the joints being closed with either copper lacquered bands or black japanned bands, as desired.

Air-cell insulation is readily obtained in sheets and blocks for application to ducts and flat surfaces.

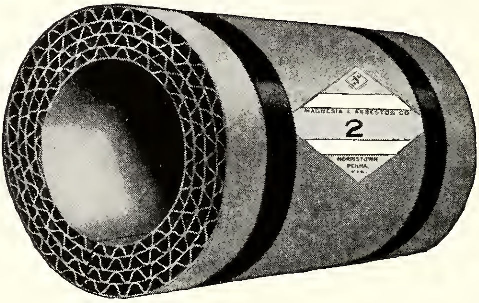


Fig. 17.2. Section view of air-cell insulation. (*Courtesy of Norristown Magnesia & Asbestos Co.*)

Asbestos Paper. The wrapping of warm air ducts with asbestos paper is a common practice, especially where the ducts ascend from floor to floor inside walls or partitions, and where the thickness of air-cell or other insulation is too great to be accommodated.

Asbestos paper does not have a great deal of value as an insulation, because of its tenuity, but its quality of being fire-retardant has some worth where the hot duct traverses combustible materials. For this reason, we find it in many standard construction specifications.

Standard rolls are available in 18-, 24-, and 36-in. widths, weighing 50 or 100 lb. The lightest weight paper is 0.015 in. thick, and the heaviest $\frac{1}{16}$ in. with four or five intermediate thicknesses generally available. A 100-lb roll of the 0.015-in. paper will contain a total of approximately 1,725 sq ft, the $\frac{1}{16}$-in. paper, 381 sq ft.

Asbestos Millboard. We often use asbestos millboard for giving fire protection to the ceiling immediately above furnaces and boilers. Although it does possess insulating value (the value of asbestos is approximately 0.80 at 122 F), we think of it mainly as a boiler-room structural accessory.

Standard boards are 42 by 48 in. in area, and vary in thickness from $\frac{1}{16}$ to $\frac{1}{2}$ in. The thinnest weighs approximately $\frac{1}{3}$ psf, the $\frac{1}{4}$-in. boards, 1.00 psf, and the $\frac{1}{2}$-in. material, 2.00 psf. Manufacturing variations are reflected in a plus or minus allowance of 10 per cent against these weights.

The Glass Insulations. During the past few years, improvements in glass making and drawing have resulted in the use of glass fibers for insulation, now widely accepted for low-temperature application. As with the other insulating materials, we can obtain spun glass and foam glass in preformed shapes, blocks, and blankets.

This material has a low *k* factor, around 0.25, and is light in weight, averaging about 1.00 lb per cu ft, in the spun fiber form. For foam-glass blocks, the density increases to an average of 10½ lb per cu ft, and the conductivity reflects this increased density by jumping to 0.55 at 300 F.

Foam-glass blocks are practically nonhygroscopic, but some care must be exercised in selecting the spun fiber insulations as these have a tendency, unless specially sealed, toward picking up moisture from the air.

The foam blocks are notably suitable for insulating tanks and roofs and for providing vapor and moisture seals in the walls of buildings. They come in slabs, 12 by 18 in. and from 2 to 5 in. thick.

If we wish to use the fiber glass for our insulation, we can choose from batts, rolls, blankets, bulk, or preformed shapes. While some of these forms are suitable for service on temperatures up to 1000 F, we should clear each proposed application with the manufacturer before making a decision. Some forms of this material irritate the hands of the applicators; consequently the wearing of gloves should be specified as a general precautionary measure.

Vermiculite Concrete. Here is a material winning increasing acceptance by architects and construction engineers, primarily because of its light weight and good strength, as compared with conventional concrete.

Vermiculite is a mica and portland cement aggregate, processed at a temperature of 2000 F. It is noncombustible and nonhygroscopic.

Commercial densities are normally three in number: 20, 24, and 34 lb per cu ft, and these densities offer *k* factors of 0.60, 0.65, and 0.79, respectively, at 70 F.

Although vermiculite could be used for low-temperature insulation of ducts, most of our encounters with the material will be in the structural field, where we must compute the heat loss through vermiculite floors and ceilings.

Corkboard. Corkboard is manufactured by compressing the granules of shredded cork, generally by steam, the natural resins of the material being the only binder needed.

This material is best known as a cold insulator, usually found in cold storage and refrigeration installations. However, recent applications beneath the pipe coils of radiant heating systems appear to be delivering satisfactory service in arresting the escape of heat downward into the earth. We can expect to see more pioneering with cork in this field.

Cork blocks are furnished in standard dimensions of 12 by 36 in., 24 by 36 in., and 36 in. square. The usual thickness is 2 in., which is being used under the radiant floors, but the material is available on request in any thickness from 1 to 6 in.

The *k* factor of cork is 0.30 at ordinary room temperatures, usually about 70 F.

Cork will burn, and prior to burning, chars, giving off a dense and choking smoke throughout the process. If we must have noncombustibility, cork is not for us. The manufacturers of corkboard are experimenting with a process that will make this insulation noncombustible, or at least fire-resistant, but no public announcements have been made as yet concerning either the success or abandonment of the project.

Cork weighs approximately 5 lb per cu ft.

Aluminum. This metal insulation operates on the reflection principle, heat rays rebounding from the bright surface, rather than pursuing the slow penetration which is characteristic of other insulators.

The first applications simply employed crumpled aluminum sheets, or aluminum foil; the latest design resembles somewhat the air-cell section, except that aluminum is used to replace some of the corrugated asbestos paper, as shown in the cross section (Fig. 17.3).

This material is supplied in accordionlike sheets, and a standard shipping carton 39 in. long by 15 in. wide and 6 in. thick carries 500 sq ft, with a net weight of 39 lb, or 1¼ oz per sq ft.

The heat conductance factors are 0.044 downward, 0.08 upward, and 0.073 wall heat.

When spread out across joists to its normal thickness, its over-all depth is 2½ in.

Aluminum is not combustible, but it does melt, and this insulating form is reported as melting at 1250 F. There is no deterioration from moisture, since the water permeability is zero.

Wood-cellulose Blankets. These materials present a good *k* factor, averaging 0.27 for a standard thickness of 1 in. In addition, they can be used to achieve a vapor seal, and possess definite sound-deadening properties.

They are purchased in standard rolls of 16, 20, 24, and 48 in. wide, the first three widths containing 200 sq ft of standard thick material, the largest width, 500 sq ft. We can expect a square foot of the standard thickness to weigh 0.12 lb.

Two other thicknesses are available, the commercial weight, only ½ in.

thick, and the double standard of 2 in. The weight per square foot is affected accordingly, and so is the *k* factor.

These blankets, generally recommended for warm air ducts and similar flat surfaces of low temperature, are slow burning, consequently we cannot use them if fireproofing is essential. Another feature that deserves some attention is their susceptibility to sloppy installation. A careless or hurried craftsman can produce an almost incredibly bad appearance, with the insulation looking as though it had been thrown, rather than fitted, into place.

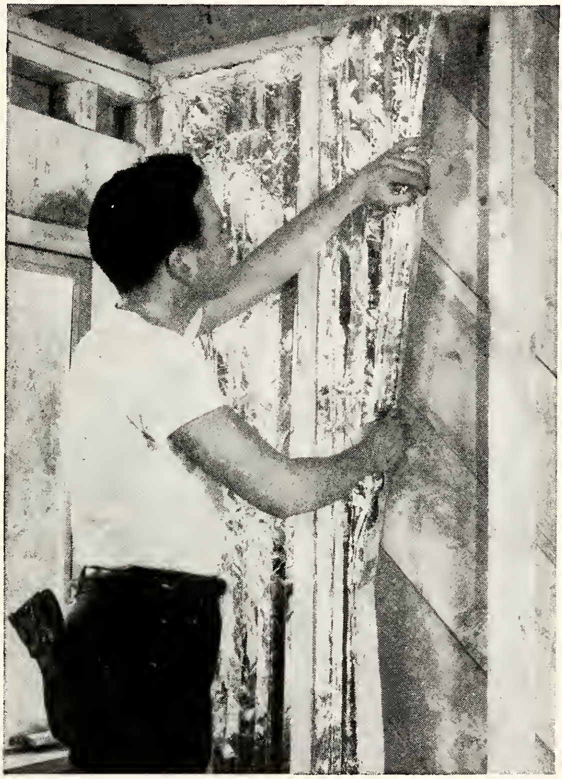


Fig. 17.3. Aluminum insulation being installed in a wall. (*Courtesy of Infra Insulation, Inc.*)

High-temperature Combinations. When insulating high-temperature surfaces, whether on pipes, breechings, or ducts, the common practice is to place a single layer of high-temperature material against the surface, followed by one or more additional layers of the cheaper 85% magnesia, or asbestos felt.

Table 17.4 indicates the appropriate thicknesses for flat surfaces and Table 17.5 the thicknesses for pipes.

TABLE 17.4. THICKNESSES OF INSULATION FOR FLAT SURFACES

Temperature of hot surface, F	Thickness, in.		Total thickness, in.
	High-temp. blocks	Secondary blocks	
600– 700	1½	2½	4
700– 800	2	2	4
800– 900	2	2	4
900–1000	2½	2	4½
1000–1200	3½	1½	5
1200–1400	3½	2	5½

TABLE 17.5. THICKNESSES OF INSULATION FOR PIPES

Pipe size, in.	Thickness, in.				
	High-temp. blocks	Secondary blocks		High-temp. block	Secondary blocks
	600–800 F,	700 F	800 F	800–1000 F	
Up to 1½	2	None	None	2	None
2	1¼	None	None	2⅛	None
3	1⁹⁄₁₆	None	None	2¹⁄₁₆	None
4	1⁹⁄₁₆	None	None	2¹⁄₁₆	None
6	1½	1½	2	2¹⁄₁₆	1½
8	1½	2	2½	2	2
10	1⁹⁄₁₆	2	2½	2⅛	2
12	1⁹⁄₁₆	2	2½	2⅛	2
14 and larger	1½	2	2½	2	2

Insulating Plaster. Gypsum plaster may be mixed with vermiculite aggregate, thereby achieving a lightweight plaster having approximately four times the insulating value of plaster formed with conventional sand. The difference in weight is represented by the difference between the vermiculite density of 8 lb per cu ft and the 100 lb for sand. We can assume a k factor for this plaster of 0.85.

If insulating plaster is used, we can run higher temperatures in a radiant ceiling coil without encountering any increased danger from cracking. This plaster is fireproof, and, when mixed as a special plastic by the manufacturer, has sound-absorbing characteristics as well. The shape of the labyrinthine voids is such that sound waves are trapped therein and dissipate most of their energy within the material.

Insulating Cements. For sealing up the joints of our applied insulation, we should use the cements furnished specifically for this purpose by the manufacturer of the insulation.

These cements, of mica and asbestos, are good for temperatures all the way up to 1900 F; when made of magnesia, the limit is 600 F, for asbestos, 1000 F. For the foam-glass blocks, the manufacturer customarily furnishes either asphalt or pitch, which is mopped on in the conventional fashion.

Cork joints usually are set up in a hot bitumastic, and are mopped with the same material for a finish coat.

Special Application Problems

Irregular Surfaces. Oftimes we are faced with the need to insulate hot surfaces, whose irregular contours make their adequate and neat covering something of a problem. In these cases, we should proceed in any one of the following manners:

1. We can form cables by twisting together two strands of No. 16 BWG black annealed iron wires, and then attach these cables by spot welding directly to the hot surface. The insulating blocks are then wired securely to the cables. This arrangement does not incorporate an air space between the hot surface and the insulation.
2. Instead of cables, we can use steel rods which are given offset bends before welding to the surface, thereby providing an air space under the insulation.
3. For spanning ribbed surfaces, we can apply wire mesh and attach the insulation to the mesh. Alternatively, expanded metal lath may be used.

After the insulating block is in place, a good plan is to reinforce the job with poultry netting, and then trowel on a ¼-in. coat of insulating plastic cement, which achieves for us a smooth, hard, and finished surface.

Insulating Breechings. For small breechings in residential installations, where stack temperatures normally are low, if we cover the breeching with 1½ in. of 85% magnesia, or asbestos felt, and surface it off with a hard finish of magnesia cement, ¼ in. thick, the job will be eminently satisfactory.

Large breechings, often encountered in the boiler plant that serves an extensive community or must carry a heavy load, tend to run to high breeching temperatures, 700 to 800 F being not extraordinary. These temperatures of course reflect inefficient operation or an overloaded plant, but they do occur; and we should consider their possibility when designing the insulation.

On these large breechings, an air space is desirable between the metal and the insulation, at least 1 in. of clearance being the minimum recommended. This space can be obtained by using offset rods welded to the breeching or by the attachment of expanded metal lath.

After the insulation is affixed in place, we apply a ¼-in. layer of appropriate cement, trowel it hard and smooth, and then wrap the entire breeching in an 8-oz canvas jacket. The jacket is pasted in place save for the abutting edges, which should be sewed with at least three blind stitches to the linear inch.

Asphalt Roofing. The sort of asphalt roofing we want for weather protecting our outside insulated piping should meet these specifications:

Avg. gross weight..........	½ psf
No. of plies................	3
Flexibility.................	At a temperature of 77 F should stand bending through 180 deg over a ⅖-in. mandrel, without showing any cracks
Efficiency of bond..........	At a test temperature of 176 F, the roofing should give no evidence of sliding, blistering, or absorption of the surface coating

Refractories

What Is a Refractory? A refractory is a material that under impact from a light or heat ray changes the direction of the ray. By another definition, a refractory is a material that resists reduction, usually by heat, as ore is reduced.

The refractories that concern the heating engineer actually meet both of these definitions. By withstanding the heat of the furnace, they maintain their existence, and while they exist, divert and reflect and refract the heat rays from the burning fuel, thereby locking up the heat of combustion in the furnace, where its presence accomplishes the most good.

How Refractories Are Made. The basic material of rock, clay, or suitable ore is shaped for our use by one of the following methods:

1. By hand working which is a comparatively slow and expensive process.
2. By the extrusion method, in which the plastic material is forced through a die that gives the bricks the desired width and thickness, the individual lengths being sawed off as the conformation leaves the die.
3. By means of a power press, which squeezes the stiff, premixed mud into the desired shape.

Probably the most common of the refractories is the familiar fire clay, which consists of mineral crystals held together by a glasslike bond. Other materials that go into our furnace blocks are silica and alumina.

The fire clays weigh from 120 to 150 lb per cu ft, the silica bricks, 102 lb, the aluminas from 136 to 153 lb, depending on the precentage of alumina present. Increasing the alumina increases the weight.

Desirable Features of Refractories. To discharge its duty efficiently, a refractory should possess the following qualities:

1. It should resist successfully both erosion and corrosion.
2. It must withstand thermal stresses at high temperatures.
3. It must withstand the physical stresses that result from its position in a furnace wall.
4. It must reflect and refract heat, resisting the passage of heat through its body, just as an insulation does. Unfortunately, the considerable density needed to meet the other requirements of a good brick lowers its insulating value; consequently we frequently back up the refractory furnace wall with a second wall of insulating brick. These latter are firebrick too, but in their manufacture they have been provided with a multiplicity of air voids, and their density sometimes amounts to no more than 18 lb per cu ft.
5. The brick should resist penetration by slag. Slag destroys the physical and thermal virtues of the brick, as the direct result of a chemical reaction.

Specifying Refractories. Most of the residential and small industrial boilers are delivered to the job with their refractories either in place or in a box accompanying the shipment. Large plants, however, present another aspect. Here the purchaser is generally expected to specify the refractory wanted, setting forth the characteristics he feels are essential to satisfactory operation. For these large installations, here are the items we should include in our tabulation of wants:

1. Specify the fuel to be used. If this is coal, endeavor to supply information on the fusing temperatures of the ash, since this characteristic will affect the life of the refractory. Low fusing temperatures, below 2200 F, will produce early penetration and deterioration of the brick. Moderate fusing temperatures are considered to fall between 2200 and 2600 F, and above this, the term used is refractory. Low-porosity brick is desirable for any ash that fuses at low temperatures.
2. Designate the type of brick wanted and its pyrometric cone equivalent. The PCE of any brick is determined by the manufacturer in a test oven. A cone of the brick being tested, of the same size as a standard Segar cone, is set in the oven with several Segar cones, and when the sample softens and bends, the Segar cone which shows the same degree of softening and bending indicates the PCE of the sample.

The Segar cone originated in Germany, and although the name is rather loosely used throughout the trade, the American version has somewhat different values, and is calibrated by the American Society for Testing Materials under the name of Standard American Pyrometric Cones. The latter system of ratings is the one commonly used, regardless of the name it goes by. Some representative values are shown in Table 17.6

Table 17.6. Representative Values of Standard American Pyrometric Cones

Cone No.	*Temperature, F*	*Cone No.*	*Temperature, F*
05	1904	15	2615
01	2093	20	2786
1	2120	30	3002
2	2129	34	3200
10	2381	38	3335

In specifying the material we want, we shall find fire clay and silica the lowest in price, and in many ways the easiest to handle. The PCE of these bricks, together with the alumina, and highly refractive magnesia and chrome bricks are given in Table 17.7.

3. A problem in furnace construction that we must solve concerns the expansion and contraction of the brick. Fire-clay and silica brick expand at high temperatures, while alumina brick tends to shrink after a time. We might keep these changes within reason if we ask for a brick that does not contract more than 1½ per cent, or expand more than 1 per cent of its original length, when tested for 5 hr at 2552 F, in accordance with the applicable ASTM directives.

Table 17.7. Typical Pyrometric Cone Equivalents of Firebrick

Brick	Standard pyrometric cone	Temperature, F
Fire clays:		
Superduty	33–34	3173–3200
High-heat duty	31–33	3056–3173
Intermediate duty	28–31	2939–3056
Moderate duty	26–28	2903–2939
Low-heat duty	19–26	2768–2903
High-alumina bricks:		
50% Al_2O_3	34–35	3200–3245
60% Al_2O_3	36–37	3290–3308
70% Al_2O_3	38	3335
80% Al_2O_3	39	3389
Silica brick	31–32	3056–3092
Magnesia brick	Over 38	Over 3335
Chrome brick	Over 38	Over 3335

4. Rapid heating and cooling are likely to create cracks and spalling in any brick. Superduty fire-clay brick offer the greatest resistance to spalling, with silica brick probably coming second. The very hard magnesite and chrome refractories are less resistant to spalling than are the other materials. If we are worried about spalling, that is, if the boiler furnaces may be completely shut down over week ends, resulting in periodic heating and cooling, fire-clay and silica brick are probably best suited to our requirements. Both of these materials, however, soften and deform at lower temperatures than do the alumina bricks, and if the fire-clay or silica walls are yielding short service, we will find better economy in high alumina replacements, despite the lesser resistance to spalling of these latter.

Standard Sizes and Forms. The names and dimensions of firebrick commonly encountered in boiler work are given in Table 17.8.

TABLE 17.8. STANDARD DIMENSIONS OF FIREBRICK*

Name of Shape	*Dimensions, In.*
9-in. straight	9 × 4½ × 2½
Small 9-in. brick	9 × 3½ × 2½
9-in. soap	9 × 2¼ × 2½
9-in. checker	9 × 2¾ × 2¾
9-in. split	9 × 4½ × 1¼
9-in.–2-in. brick	9 × 4½ × 2
9-in. No. 1 arch	9 × 4½ × (2½–2⅛)
9-in. No. 2 arch	9 × 4½ × (2½–1¾)
9-in. No. 3 arch	9 × 4½ × (2½–1)
9-in. No. 1 wedge	9 × 4½ × (2½–1⅞)
9-in. No. 1 key	9 × (4½–4) × 2½
9-in. feather edge	9 × 4½ × (2½–⅛)

* *Note:* For 9-in. circle brick, jamb brick, skew brick, and other complex shapes, refer to a refractory catalogue. A series of bricks as given in this table, but starting with a thickness of 3-in. rather than 2½-in., also is available.

Castable Refractories and Ramming Mixes. Originally, a castable refractory was a material that we poured into place between forms, as concrete is poured. Like concrete, it hardened over a period of time.

In modern practice, the term "castable" is being used somewhat interchangeably with "ramming mix" and "plastic." The reason is that while ramming mixes were pounded into place by our fathers and by many of our contemporaries we actually can apply the stuff by means of a gun, water being introduced at the nozzle, and the mixture then being projected home by pneumatic pressure.

The current castables and ramming mixes are suitable for temperatures up to 3000 F when applied by gunning and used for patching. When cast into form, duty should be limited to temperatures between 2400 F and 2800 F, as recommended by the manufacturer.

The density of these materials varies from perhaps 75 lb per cu ft up to 155 lb, the density increasing as the duty temperature increases.

Some Notes on Furnace Design

A furnace that is designed to fit the flame, and at the same time provides ample volume for proper mixing of air and fuel, is likely to deliver the most satisfactory results. In large installations, considerable thought is given to these factors by the designing engineer, and all the essential calculations are made, but oftimes the residential and small industrial boilers, if the furnace is not provided by the boiler manufacturer, receive short shrift. Any arrangement of firebrick may be judged acceptable by the inexperienced installer, with trouble as the main dividend.

TABLE 17.9. FURNACE DIMENSIONS FOR DOMESTIC OIL BURNERS

Fuel rate, gal/hr	Center line, nozzle to floor, in.	Furnace width, in.	Furnace length, in.	Wall height, in.
1.0	6	10½	12	14
1.3	6	11½	13½	16
1.7	6½	12	15	16
2.0	6½	13	16	16
2.5	7	14	17	16
3.0	7	16	19	18
4.0	7½	17½	20½	18
5.0	7½	19	22	20

TABLE 17.10. FURNACE DIMENSIONS FOR GAS-OIL BURNERS

Fuel consumed		Center line, burner nozzle to floor, in.	Furnace width, in.	Furnace depth, in.	Wall height, in.
Oil, gal/hr	Gas, cu ft/hr				
6	900	14½	28	56	44
9	1,350	14½	34	62	44
11	1,650	17½	39	74	49
15	2,250	19½	45	95	56
18	2,700	21	55	104	57

As a guide to the design of these smaller furnaces, the data in Table 17.9 have been compiled from several sources, largely from the "Babcock and Wilcox Firebox Handbook."

When we plan to use a combination gas-oil firing arrangement, the dimensions will vary as required by the characteristics of the burners selected. Table 17.10 shows some of the minimum dimensions recommended by one manufacturer of these combination burners; they are suitable for boilers handling rather sizable loads, up to about 8,400 sq ft of equivalent direct steam radiation.

These tables give average data. The sensible, prudent rule is to check our furnace dimensions, as we project them, with the manufacturer of the burner before actual construction is undertaken.

For small furnaces, corbeling of the bridge wall inward will sometimes help to produce more efficient combustion. This corbeling need not be bold, just enough to give the combustion gases an upward sweep rather a wiping progress across the top of the wall.

Questions

1. Enumerate the six marks of a good insulation.
2. Where would you use rock wool? Why?
3. What is asbestos sponge felt? What is its top temperature?
4. What is 85% magnesia?
5. What added precautions should we take for outdoor service?
6. Where do we use asbestos air-cell insulation, and why?
7. Describe the characteristics of the glass insulations.
8. How does aluminum insulation differ in principle from the other types?
9. How would you insulate the breeching between your own heating boiler and chimney?
10. What is the purpose of a refractory?
11. Name the four items of basic information that should be given in a refractory specification.
12. Where would you use castable refractories or a ramming mix? How would you apply the material?
13. What is meant by the "pyrometric cone equivalent"?
14. What kind of refractory would you use for building a new furnace in your own boiler? How would you handle repairs to the existing furnace brick?
15. Where do we use insulating firebrick? How does it differ from other firebrick?
16. What is the relation between density and the heat-transmitting characteristics of firebrick?

CHAPTER 18

FIREPLACE DESIGN AND PERFORMANCE

Principles of Design

Why Have a Fireplace? This is not a foolish question as might be supposed, since fireplaces do, as of this modern day, still provide the major source of heat for comfort and cooking in many areas of the world.

In our Southern states, the building of a new house with only a fireplace for winter heating is quite common. Below freezing weather is not unknown there either, and may occur several times between autumn and spring, but the fireplace is expected to meet the situation nicely.

With the advent of steam, hot water, and warm air heating, and the growing popularity of these good providers of winter comfort, the construction of fireplaces declined; but the remembered cheer and picturesqueness that are part of an open fire, and the distinction that it adds to a room, have brought fireplaces back to the home. Whether we design a fireplace for heat or distinction, we must learn to design it correctly.

Basic Rules of Good Design. A fireplace is composed of six units: the opening, the back wall, the side walls, the throat, the smoke chamber, and the flue. If we fail to provide any one of these units with its correct relationship to the others, unsatisfactory performance of the fireplace may be our penalty.

Here are basic rules of design affecting each unit:

1. *The opening.* The opening is what we see when we look at a fireplace. We can spoil the working efficiency of the opening by making it too wide or too high. Table 18.1 offers sound dimensions for the opening, these ratios having been proved by their satisfactory performance. Generally the height of the opening suited for most homes is seldom more than 30 in., with the width slightly greater.

2. *The back wall.* Table 18.1 indicates more or less standard distances from the front edges of the side-wall brick (not the mantel) to the back wall, measured on the floor of the fireplace. In 1796, the English scientist Count Rumford made lengthy and exhaustive studies of fireplace design, and as a result promulgated most of the rules we follow today.

One of these affected the back wall. Count Rumford recommended that it be sloped forward to augment the radiation of heat into the room and to direct the products of combustion forward to a throat located immediately in back of the mantel.

This slope should begin with the third course of firebrick above the floor.

3. *The side walls.* Side walls that are set perpendicular to the back wall tend to bottle up the heat rather than to reflect it outward, as a good wall should do. Consequently, the side walls should be set at an angle, determined largely by the general appearance of the fireplace, but usually somewhere around 30 deg.

4. *The throat.* Here is a critical unit of the fireplace. If it is too small, the fire will smoke. If it is too large, the draft will languish. If it is not in the right place, the effect of the fireplace is diminished.

Table 18.1. Standard Fireplace Dimensions

Finished opening, in.			Flue lining, in.	
Width	Height	Depth	Rectangular	Round*
24	28	16	8½– 8½	10
28	28	16	8½–13	10
30	30	16	8½–13	12
34	30	16	8½–13	12
36	30	16	13 –13	12
40	30	16	13 –13	15
42	30	16	13 –13	15
48	33	18	13 –13	15
54	36	20	13 –18	18
60	39	22	18 –18	18
72	40	22	18 –18	18

* *Note:* Round flues tend to be a little more efficient than those of rectangular cross section.

The throat should extend across the full width of the fireplace and be equal in area to at least 15 per cent of the fireplace opening. Its best position is well forward, directly behind the mantel, and 4 in. above the lintel. The lintel, incidentally, is the cross member at the top of the fireplace opening. In colonial days, the lintel was frequently a heavy oak timber, a material capable of withstanding successfully considerable heat. In present practice, it is likely to be formed of iron, and is sometimes a part of the damper frame.

The lintel depth is usually the depth of the brick between the throat and the face of the mantel, a dimension that preferably is 8 in.

5. *The smoke chamber.* As shown on the sketch (Fig. 18.1) the smoke chamber is a space, or compartment, that connects the throat to the flue.

In the smoke chamber the gases of combustion are brought together for discharge through the flue. The back of the smoke chamber floor is a more or less horizontal shelf, intended to block downdrafts and, in cooperation with the damper, to redirect these drafts right back in the direction whence they come.

Unfortunately, the damper does not invariably cooperate, as the result, in some instances, of the damper being pivoted on a longitudinal center line, whereas all properly designed dampers pivot along the longitudinal back edge. Dampers incorrectly pivoted should not be accepted by the designer.

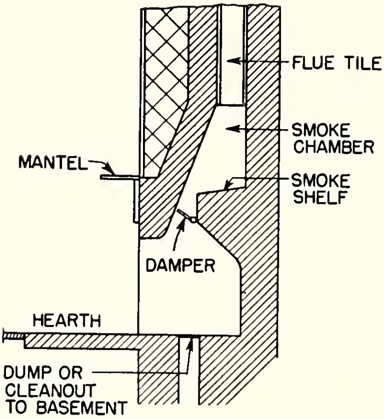


Fig. 18.1. Smoke chamber for an open fireplace.

The smoke chamber should be designed for the full width of the throat and should be sufficiently deep to place its back wall flush with the back wall of the flue. The depth of the smoke shelf is therefore the distance between the throat and the back wall.

The side walls of the smoke chamber should slope, like the sides of a pyramid, to meet the flue. This slope must be not less than 45 deg, and 60 deg is good. In any event, the flue must be centered above the width of the fireplace, and any offset found unavoidable should be accomplished somewhere higher than the point of junction between the smoke chamber and the flue.

6. *The fireplace flue.* For each fireplace there is a best size of flue. These sizes, both rectangular and round, are shown in Table 18.1.

In the absence of any better guide, a rule of thumb for flue sizes calls for a flue area equal to 10 per cent of the fireplace opening. Thus if we had a fireplace, say, 36 by 30 in., the flue area should be at least 10 per cent of this area, or 108 sq in. Compare this with the size recommended in the table, and the rule of thumb appears to be a little tight. The table calls for a flue lining of 13 by 13 in., but these are outside dimensions, and the actual inside area is 126.5 sq in. The 12-in. round flue, also recommended, develops an area somewhat closer to the rule-of-thumb result, being rated at 113 sq in. In the circumstances, and faced with this evidence, it is suggested that we modify the rule of thumb to a basis of

11 per cent, since the 10 per cent formula looks as though it should be increased by 10 per cent of itself.

Offsets in the flue should be kept as close to the vertical as practicable. A horizontal offset, for example, is a poor one, because the horizontal run increases flue friction but does not contribute any change of elevation which might augment the draft and thereby cancel out to some extent its imposed frictional burden.

In fireplaces of Shakespeare's day, the chimney often contained from four to six flues, each of these flues being assigned to a separate fireplace, usually in various adjoining rooms. Present-day chimneys seldom contain more than two flues, one for the heating plant, and one for the fireplace; but if we are not careful these two flues can cause trouble quite equal perhaps to all six in Shakespeare's arrangement.

In the first place, the tile liners of flues, when located in the same chimney, should be separated by at least 4 in. of brick. In the second place, the liner joints should be staggered, particularly if the liners are placed against each other, without the separating partition of brick.

The objective of all this care is to avoid cross leakage, for if this condition develops, both of the involved flues will fail in their respective duties.

The areas of flue liners in square inches appear in Table 18.2.

Table 18.2. Standard Sizes of Flue Tiles

Rectangular dimensions, in.	Flue-tile area, sq in.	Round flue tiles	
		Diameter, in.	Area, sq in.
8½– 8½	52.5	10	78.5
8½–13	80.5	12	113.0
8½–18	109.6	15	176.7
13 –13	126.5	18	254.5
13 –18	182.8	20	314.1
18 –18	248.0	22	380.1

A Late British Design. Considering that fireplaces are still in service for much of England's heating and that most of what we know of fireplace design originated with an English scientist, we need not be surprised that the British have produced a new form of fireplace which reportedly doubles or nearly doubles the approximately 10 per cent efficiency characteristic of current design.

In this new design, which has been reported in some detail by the University of Illinois Small Homes Council, the back hearth, or floor under the firedogs, is depressed; next the damper has been omitted, the horizontal smoke shelf suffers the same fate, and the lintel has been smoothly rounded off. The sketch (Fig. 18.2) indicates the general trend of this design.

The whole arrangement seems quite sensible to this writer, particularly in view of troubles experienced with both dampers and the smoke shelf. Complete removal of an existing damper has improved the performance of many a fireplace that was unsatisfactory before.

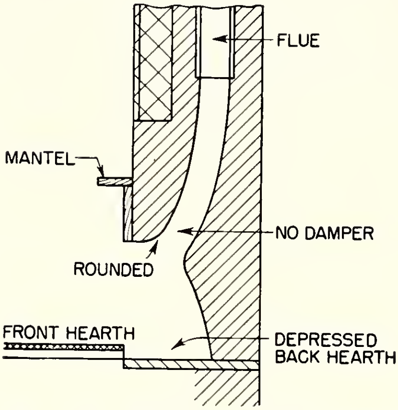


Fig. 18.2. Recent design of English fireplace.

Problems of Performance

How to Light a Fireplace. In these civilized days, fireplace complaints often originate from owners who simply do not understand the fine points of kindling an open fire.

If we lay a single log on the firedogs, heap some paper or shavings underneath it, and then expect by striking a match to achieve a roaring open fire, we are naïve, but not singular. This approach to the problem is being practiced every day, and the results invariably present a fixed pattern: the log does not burn.

The fire should be started with a few light sticks of dry wood, crisscrossed on the firedogs. Such a pyre can be lighted almost with a single match, and at worst a bit or two of paper is ample. Alternatively, we can use a hearth torch, which consists of a porous stone soaked in kerosene from a closed iron pot kept near the fireplace for convenience. The blazing stone, lying beneath the crossed sticks, soon has them crackling merrily.

We now add heavier wood to the pile on the firedogs. Two small- to moderate-sized logs may be laid side by side, with a third arranged on top to form a pyramid. Through the interstices of these logs the flames find drafty channels, the logs ignite, and the fire is now ready to continue under its own power.

The starting rule is three logs. Later, when the fireplace is throughly heat-soaked, a single large timber may smoulder effectively for hours, giving off a strong heat until its entire substance becomes ash. But at the start, a single tree trunk and a match simply refuse to cooperate. Remember the rule.

Smoky Fires. Count Rumford discovered that the relationship between the position of the throat and the position of the lintel could be a cause of smoke. The lintel, as we have noted before, should be about 4 in. below the throat opening for best operation. This cause should be eliminated before it occurs; that is, we should design our fireplaces properly before they are built.

Inadequate draft and downdrafts likewise create a smoky atmosphere around the hearth. Inadequate draft is our repayment for building a chimney that is too low, or for installing too small a flue, or for missizing the throat. Not infrequently the damper turns out to be something of a misfit, and the simple removal thereof has been known to stop smoke.

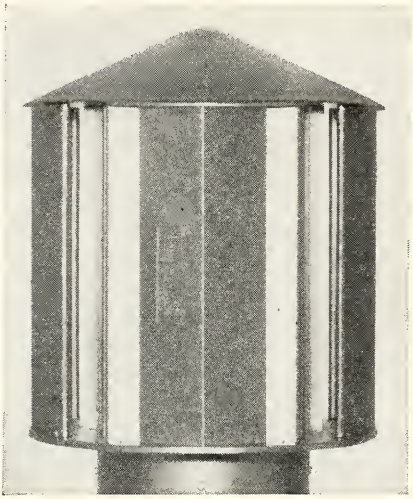


Fig. 18.3. Air exhauster for combatting downdrafts. (*Courtesy of The G. C. Breidert Co.*)

If the flue is too small, there is little we can do except rebuild the chimney. Short chimneys, however, can be raised, throats can be chipped out to size, and obstructions, which appear in flues more often than we suspect, can be removed.

Downdrafts may plague fireplace operation as the result of trees or other blanketing structures being present in the neighborhood of the chimney top. The subject of downdrafts is covered in some detail in Chap. 16 which is concerned exclusively with chimneys and flues. However, for fireplaces, Fig. 18.3 illustrates a type of exhauster that functions very satisfactorily. Whatever the direction of the wind, its energy is captured for the increasing of the fireplace draft. The appearance of the exhauster is an improvement over the tall stovepipe that at times is observed disfiguring an otherwise attractive residence, and the cost is likely to be less than is incurred by the extension of a brick or stone chimney.

The Open-hearth Fire. Figure 18.4 illustrates the type of open fire that occasionally contributes a dramatic appearance to large halls, such as hotel lounges and clubs. The smoke hood and smokepipe, sometimes suspended from the ceiling, may be decorated rather ornately, perhaps to suggest the costly and luxurious appointments of a sultan's palace.

The value of this fireplace is strictly beauty; as a dispenser of heat it is comparable to a campfire of equivalent size kindled on the plains. There is no brick wall to reflect the heat, and no draft of any consequence to whip up the embers, since the air movement tends to be slowed down by the vast open spaces between the hearth and the smoke hood.

A modified open hearth is sometimes built into homes, open on two or three sides, and is intended to permit a view of the fire from two adjacent rooms or areas. As comfort heating equipment, it is an improvement over the full open hearth to the extent of possessing at least one reflecting wall of brick. Just how important this wall is, we can estimate by recalling that the best of our present fireplaces delivers effectively to the room only 10 per cent of the heat of combustion, using all three walls for this purpose. Clearly we cannot expect too much from a single wall.

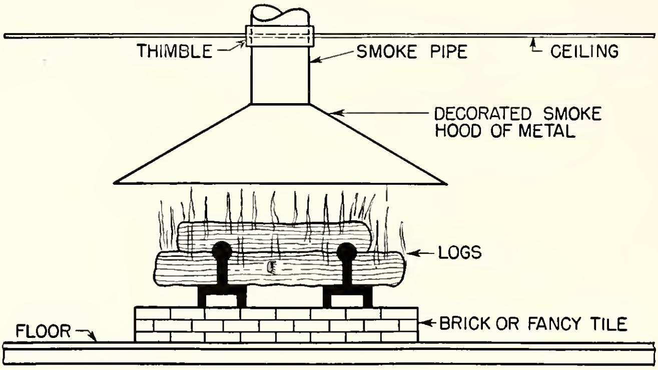


Fig. 18.4. Open-hearth type of fireplace.

Fireplace Circulators

Fireplace circulators are not new. Benjamin Franklin built a metal stove into a fireplace opening, so arranging his contraption that air from the room was coaxed into the iron passages by natural convection, and having been warmed there, returned to the room with a burden of comfort-giving heat.

Current designs employ not a stove, but a fireplace which is built of metal, so that the back and side walls reflect heat into the room in the same way as in the case of a conventional fireplace, and also simultaneously warm the room air by convection, the air being passed over the back and sides of the fireplace, either by natural draft or by fans mounted on the inlet grilles.

Figures 18.5 and 18.6 picture a fireplace of this kind. Both the air inlet and outlet connections can be served by short ducts, if desired, thus permitting the grilles to be located in the walls and surbasing to best advantage.

These metal fireplaces are readily blended into the architectural requirements of the room, since they may be faced with brick or stone to look exactly like any conventional fireplace. Sizes are standardized as shown in Table 18.3.

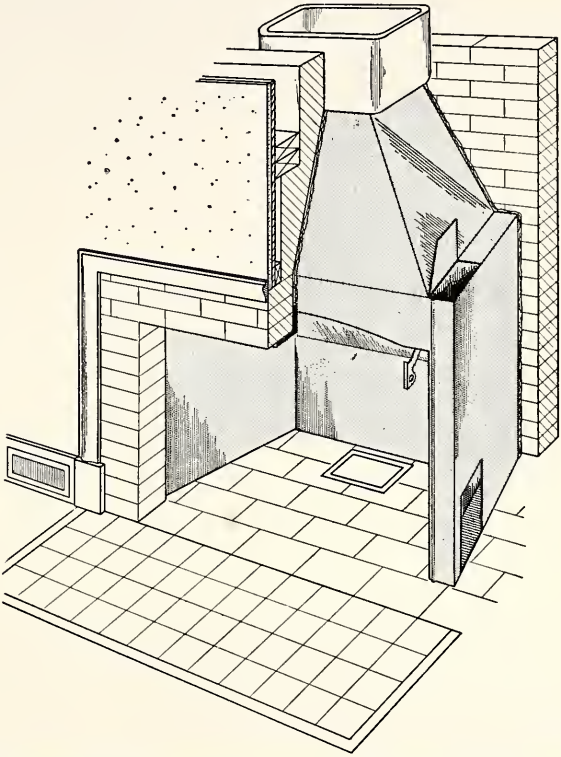


Fig. 18.5. The Heatilator for installation in open fireplaces. (*Courtesy of Heatilator, Inc.*)

TABLE 18.3. SIZES OF CIRCULATING FIREPLACES

Finished opening width, in.	Opening height, in.	Weight, lb	Flue sizes for chimneys, in.	
			Over 20 ft high	Under 20 ft high
28	24	180	8½–13	8½–13
32	26	205	8½–13	13 –13
36	28	255	13 –13	13 –13
40	30	325	13 –13	13 –18
48	32	420	13 –18	13 –18
60	36	680	13 –18	18 –18

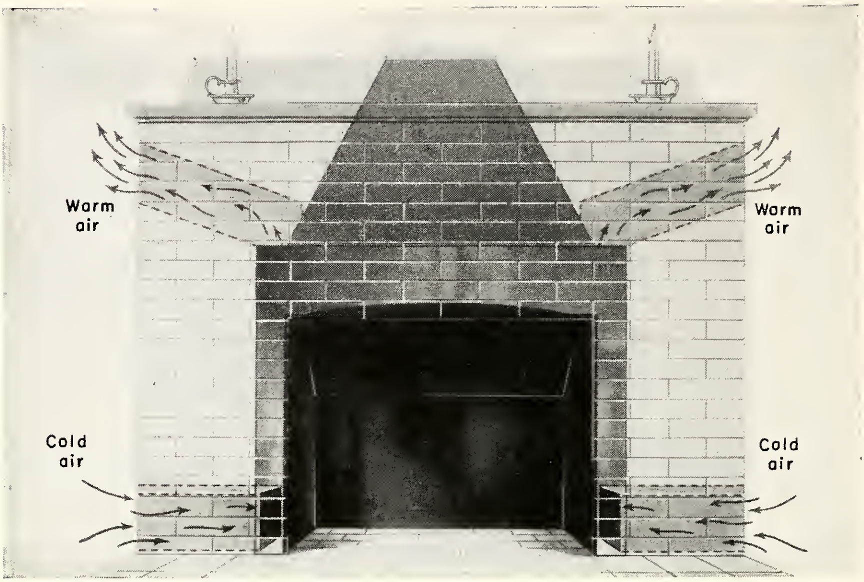


Fig. 18.6. Another view of the Heatilator.

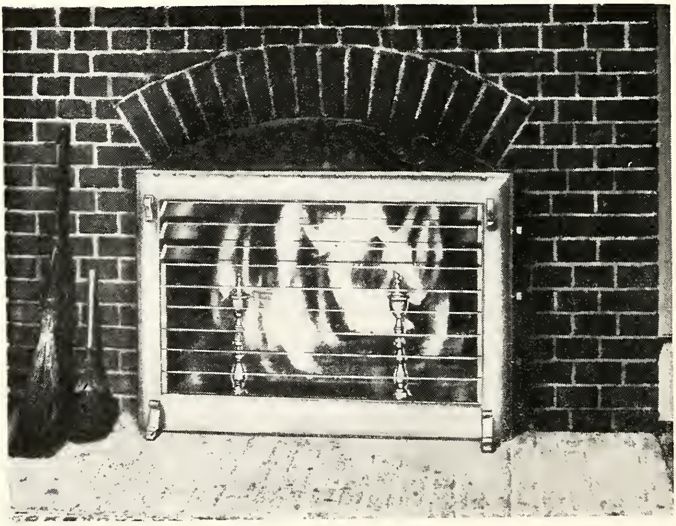


Fig. 18.7. The latest in fireplace screens. The knobs on the right control draft by manipulating the glass louvers that form the front. Air enters under the screen and leaves through the vent holes visible in the top of the device. A bronze screen behind the louvers protects the glass against sparks. (*Manufactured by the Dollinger Corp.*)

Circulating fireplaces need not be expensive. For the sizes given in the table, list prices may be expected to range from $40 to $150 each. If we use fan grilles instead of the conventional openings, these fan grilles complete will add about $16 to the cost of each inlet opening. Installation charges are additional, and must be added to complete the estimates of over-all cost.

When designing a circulating fireplace, we must be careful to provide a hearth of firebrick under the metal floor, and we must allow also for expansion. Approximately ½ in. of clearance is indicated, particularly at the corners, and this clearance should be packed with appropriate insulation.

Fireplace Screens. Normally a fireplace screen would be considered merely an accessory and would have no place in a discussion of design and practice. However, the screen shown in Fig. 18.7 actually performs draft control functions, and consequently becomes of some importance.

The screen is built of two components, the inner unit being a wire-mesh barrier suitable for arresting sparks, while the outer elements consists of adjustable glass louvers, or slats, by which the admission of air to the fireplace is governed.

The builder claims that the screen will eliminate smoking regardless of the cause, and there appears to be some merit in the claim since even downdrafts can be bottled in the fireplace simply by closing the glass louvers.

These screens currently are being offered in standard sizes ranging from 32 to 50 in. in width and from 28 to 35 in. in height.

Questions

1. Name the six basic rules of good fireplace design.
2. What is the conventional rule of thumb for sizing flues? Does our study indicate that some modification of this rule is desirable?
3. What is the fundamental rule for lighting a fireplace?
4. How would you attempt to correct a smoky fire?
5. What is the treatment for downdrafts?
6. What is the effect of the open-hearth or modified open-hearth design on fireplace efficiency?
7. If you were building a fireplace in your own home, would you use a circulating design? Why?
8. Compile a comparative cost balance between a circulating fireplace, using inlet grilles with fans, and a conventional fireplace of the same size.

CHAPTER 19

HOW TO CHOOSE A HEATING SYSTEM

First Considerations

Six Factors in Choosing a Heating System. Whether we select steam, hot water, warm air, a panel, or baseboard design may be decided, not always correctly, by the owner of a project. Actually, the heating engineer should make an analysis of each job and then, if the findings indicate that some divergence from the owner's desires would be to the latter's advantage, the original decision should be reconsidered.

We should realize from the start that most owners are nontechnical, and consequently their decisions tend to be a little unbalanced, representing a desire of the moment and usually expressed without any understanding of the facts. For example, the factor of cost may be glossed over in preliminary discussions, but after the design is completed and bids have been taken, the financial awakening may be so harsh that only an entirely different arrangement can be brought within budget limits.

Therefore, regardless of what the preliminary decisions are, the following six influences should be evaluated for each job:

1. *The Cost Factor.* On the current market and in most parts of the country, some form of warm air heating generally will be found lowest in original investment. Based on an extensive analysis of jobs by the writer, we can assign warm air a par value of 1, followed by circulating hot water at 1.5, and steam heating at 1.65.

Next, we can break down the warm air group into space heaters and duct heaters, with the former occupying the bottom of the cost ladder. In fact, we can expect to install a 50,000 Btu per hr floor heater, with outdoor oil tank of at least 100 gal capacity, complete with fuel piping and chimney breeching for less than $200.

Now let us examine some forced warm applications with ducts in various types of structures. Table 19.1 shows what we can expect to pay, per 1000 Btu per hr capacity.

This currently favorable position of the warm air cost factor will be found quite potent in changing minds and influencing people.

2. *Special Requirements Factor.* Is a large kitchen involved in the project, or is there a swimming pool to be heated, or perhaps an auditorium, either presently or in the future?

Large kitchens almost invariably require steam, not only for hot water but also for steam kettles and sometimes for ovens and coffee urns. Swimming pools impose a huge heating load on any plant, and since the most effective heat exchange is accomplished with steam, we are almost obliged to generate steam when a swimming pool is concerned. In short, substantial demands for steam provide a strong influence for designing the whole heating system to use steam.

TABLE 19.1. REPRESENTATIVE COSTS OF VARIOUS HEATING SYSTEMS

Type of installation	Type of building served	Investment cost, per M Btu*
Mechanical warm air	Residential	\$10–\$13
	Church	\$12
	Fire station	10
	School (single-story)	12
Circulating hot water	Residential	18
	Apartment (skyscraper)	13
	School (multistory)	17
	Church house	13
	Club	12
	School (single-story)	8
Steam:		
One-pipe system	School	9
Two-pipe system	School, large group	14

* *Note:* These figures have value merely as guides. Costs vary with local conditions and the general status of the labor and material markets.

Or consider a hospital. The ventilation requirements for operating rooms, as set forth by the U. S. Public Health Service, dictate a continuous supply of fresh air taken from out of doors, plus an equally continuous and positive exhaust system to complete the exchange. An arrangement of this sort almost inevitably needs ducts; therefore, would it not be economically and technically sound to use ducts throughout the hospital? Perhaps. Or possibly a steam or hot water coil in the operating-room duct circuit, with standard convectors in the nursing rooms and other facilities of the hospital, will be the best combination. Each hospital must be studied for its own peculiar conditions of layout before a decision can be made.

These illustrations point up the importance of the special requirements factor.

3. *The Summer Cooling Factor.* When summer cooling is projected at some future date for a residence, church, retail store, theater, or other comparable activity, certainly the basic installation of a warm air heating system is indicated. We size the ducts for the cooling load, since the air requirements are greater for cooling than for heating, and the owner can add the cooling equipment when convenient.

4. *The Need to Tie In with Another System.* We frequently meet projected expansions of existing buildings, a situation that poses the desirability and feasibility of tying the new heating system to the old.

Unfortunately, if the basic system was properly designed to meet the original conditions, any tie-in that is more complex than a valved interconnecting pipe, or a dampered interconnecting duct, seldom is practical. We shall need, probably, a new boiler room strategically located in the new expansion, and almost inevitably a new chimney, since chimneys are rarely overlarge. We even are quite likely to employ a new medium, perhaps steam instead of hot water, warm air instead of steam, chosen to satisfy the new demands for comfort.

Extension of an existing system is possible, however, and we are obliged in consequence to weigh this possibility before making a choice.

5. *When the Buildings Are Scattered.* Given a group of structures that are separated by streets or campus areas, we must decide if a single central heating plant, or a system of unit installations, is the more advantageous.

The central plant involves underground piping of some sort, and underground structures are expensive to install and sometimes expensive to maintain. On the other hand, the desirability of centralized control and centralized supervision becomes a powerful advocate for the single plant.

One thing is sure: Heating by ducts from a central plant is not practical when a group of buildings is involved. If we want warm air, we must go to the unit scheme, with the units either fired on the premises or served by hot water or steam from an underground distribution system.

Considerable study, especially economic study, is a must for projects of this nature.

6. *The Architectural Factor.* Warm air ducts frequently are of large size; yet on occasion can be faired into a building's walls and ceilings so successfully that architectural harmony is maintained. In these cases warm air heating involves no compromise with beauty, but if the sight of the duct is an abomination, then we must consider other mediums as our bearers of comfort.

Often we encounter the architectural necessity for running long horizontal pipes through floors and ceiling spaces, a situation that almost automatically bars steam due to the difficulties of providing adequate drainage. Water in steam pipes not only tends to proclaim its presence by a loud pounding on the iron but its bulk restricts and even stops the necessary circulation of the steam. In these cases, even though the owner wants it we should object stubbornly to the adoption of steam.

Summary by Checkoff. Whenever handling a heating job, we should begin by summarizing the important conditions on a checkoff list:

1. What are the approximate differences in cost between warm air and hot water, warm air and steam? Figure in dollars, not percentages.
2. Does the project involve any special or collateral needs for steam, for warm air, for hot water?
3. Is summer cooling a future probable?
4. Is there a possibility that future extensions of the system will be required?
5. Must we consider tying the new work to some existing system?
6. Do architectural requirements make any particular system impractical, or undesirable?

Considering System Characteristics

Mechanical Warm Air. This section gives some basic pros and cons concerning this form of heating.

In favor of this form of heating are the following features:

1. Currently, warm air offers the cheapest satisfactory installation from an investment standpoint.

2. If the architectural arrangement logically permits the running of ducts in furred ceilings, or their being disguised as pilasters or false beams, the final appearance may be more satisfactory than if we use radiators or convectors.

3. Properly designed, a warm air system is potentially an all-year conditioning system.

4. Warm air permits filtering and humidifying the air during the heating season.

This kind of heating system has the following unfavorable features:

1. Warm air with duct distribution is not suitable for heating a group of buildings from a single plant.

2. In residential installations, the sound of the fan is carried through the ducts and may be considered objectionable.

3. Cleaning or replacement of the air filters becomes a regular chore.

Circulating Hot Water. In favor of the circulating hot water system are the following facts:

1. It provides notably uniform heating, without any sudden additions or decreases of heat emission, regardless of the control cycle adopted.

2. It is currently the most popular medium for panel heating.

3. In some arrangements, notably the Trane-type of room all-year conditioner which incorporates a fan and fresh air supply to unit room convectors, as illustrated in Fig. 19.1, the hot water of winter heating is replaced by cold water for summer cooling. In this arrangement, a three-way mixing valve varies the temperature of the circulating water, both winter and summer, to meet changing demands as they occur, from hour to hour.

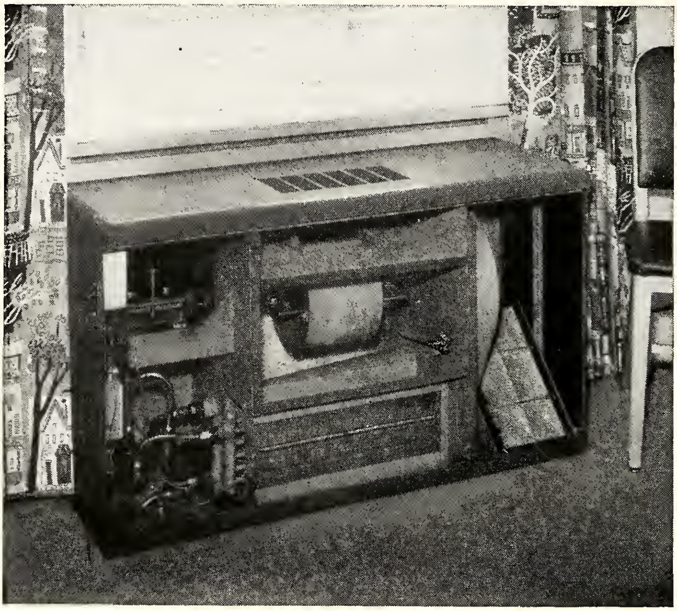


Fig. 19.1. Room all-year conditioner with cover removed showing coils, filter, and fan. (*Courtesy of The Trane Company.*)

4. Being moved by a pump, the water circulates up hill, down dale, and over the level, as necessary.

Against the system we observe that:

1. Everytime we institute a new control zone, we must contrive a piping system to serve it. Since these zone systems all originate in and return to the boiler room, we soon have a considerable physical bulk of piping to contend with.

2. Venting of the system sometimes becomes an annoying problem, particularly where long runs of level piping are involved, and vent locations are restricted.

3. Pipe friction tends to increase with time; consequently a system that has been sized rather closely to hold down investment costs may eventually become too close for comfort.

4. Balancing a circulating system, when one circulating pump serves two or more circuits, requires a good bit of cut and try. This means time, and time is money to the contractor who does the balancing.

Steam Systems. The good features of steam systems are:

1. A steam system permits fast heating and fast cooling, a notable virtue where large assemblies are concerned.
2. Along district steam lines, steam heating is a normal choice.
3. Unit heaters, radiators, and convectors may be installed in the smallest possible sizes when steam is the heating medium.
4. Since the energy of the steam provides its moving force, zone control is simplified. Zone or sectionalizing valves may be located wherever convenient.

The systems have the following complicating features:

1. Steam systems may be the most expensive to install.
2. Adequate drainage is not always easily provided. We need a pitch of ¼ in. for each 10 lin ft of steam pipe and of ½ in. for the condensate runs. Sometimes the only way to obtain this pitch is to dig pipe trenches, a costly and in some places an unsightly expedient.
3. The bulky covering needed by steam piping can introduce a space problem that conflicts with architectural treatment.
4. Boiler return machinery, such as pumps, traps, and tanks, create additional maintenance hazards.

Panel Systems. Their desirable characteristics follow:

1. All visible heating apparatus is eliminated from a room.
2. There is a possibility of reducing operating costs.
3. All floor and wall areas are freed for other use.

Their opposition points are:

1. Panels cost more to install than conventional systems.
2. Surface temperature limitations can complicate the installation by requiring coils in more than one surface.
3. The physiological reactions by room occupants are uncertain.
4. The installation is largely limited to new construction.

Baseboard Designs. Baseboard advantages are:

1. This design responds quickly to outdoor temperature changes.
2. Baseboards maintain a greater uniformity of room temperature, measured between the floor and the comfort line, 5 ft above the floor, than other types of heating, with the possible exception of a floor radiant panel.
3. Some styles are of unobtrusive appearance and detract little from the maximum floor area.

The disadvantages are:

1. Baseboards are not cheap to install.
2. If the heating medium rises above 200 F, we can expect the development of streaked walls above the baseboard.

3. When border footage is limited by the presence of doors, fireplaces, French doors, or other encroachments on wall space, installation of sufficient baseboard to meet the load requirements may become impossible. This situation is particularly critical in kitchens, where much of the wall area is blanketed by cupboards, sinks, and other culinary appurtenances.

Gas-fired Equipment for Space Heating. The desirable features follow:

1. It is comparatively inexpensive to install. For small churches, schools, and buildings of similar limited use, gas-fired unit heaters or radiators deserve the designer's careful attention. They dispense comfort on a low installation investment.

2. It is a good choice for single or isolated rooms.

It can be unattractive because:

1. In some areas, gas is an expensive fuel.

2. Burner vents are essential; otherwise the products of combustion foul the air. Vents tend to be unsightly.

3. Gas leaks can precipitate deadly explosions.

Floor Furnaces and Similar Space Heaters. The good and bad points of this type of heating are summarized in these groups:

The group of advantages shows:

1. Low cost and simplicity.

2. The floor furnaces release floor space.

The disadvantages are:

1. Heating capacity is limited. Circulation of warmed air is not practical through more than one doorway adjacent to the heater.

2. The room-type space heater occupies considerable floor area.

3. The room-type unit exposes more or less unsightly stovepipe connecting to the chimney.

4. Cold drafts along the floor are common annoyances.

5. A wide variation of temperature is produced between the floor and ceiling, as much as 20 deg.

Heat Pump. It has the following valuable characteristics:

1. It eliminates all need for a chimney and fuel.

2. A single apparatus both heats and cools, automatically switching from one cycle to the other, as demanded by the weather.

3. It reduces fire hazards, since no combustion is involved.

The less estimable features are the following:

1. It is expensive to install, and if electric rates are high, is expensive to operate.

2. Service work may be difficult to obtain in some localities.

3. If the pump is sized for the heating load, we may find it considerably larger than is necessary for cooling. This possibility complicates sizing.

Locating the Plant

Conventionally, the boiler or heater is set on the basement floor. This long-established practice has been found unsuitable, however, in some of the recently built structures, a modern trend being to eliminate the basement. As a result, we find many heaters packed neatly into closets, or safely set up in the attic.

There is no reason why a heater that operates with some form of mechanical circulation, whether water or air, cannot be located above the space being heated. A radiant ceiling, for instance, suggests a notably economical possibility if the heater is installed in the attic. With this arrangement, hot water or warm air supplied to the ceiling requires a minimum of piping or ducts.

The one-floor basementless houses are especially amenable to this treatment from the attic, even if we use the conventional warm air registers set either high or low in the walls. However, if the building is two stories or more, the operation of a warm air system will be improved by the installation of a butterfly damper in the heater discharge. Without this damper to close automatically when the heater shuts down under control of the room thermostat, the tall stacks to the registers function as chimneys, and much of the warm air in the rooms goes right back to the heater.

Steam boilers can be set on upper floors also, although this medium requires a low sump plus a condensate return pump on the bottom level to lift the returns to the boiler. Gravity circulators, whether air or water, of course are eliminated, as the law of gravity prevents the heating agents from descending willingly to lower levels. However, gravity circulations are not generally classed among modern designs, despite the tendency to use gravity warm air in extensive housing developments in order to save on plant cost.

Fire departments regard attic installations with favor, since a fire in roof areas usually is discovered early, and is comparatively slow to spread downward.

Location of the heater, or boiler, is an important matter to decide early, since the influence of location on cost is obvious.

Questions

1. Enumerate the six basic influences to be considered before making a decision.
2. Why is this analysis of applicable factors essential in every project, whether large or small?

3. Consider the heating system in your own house. Analyze it from the viewpoint of the six influences. Would you still choose the system now in place?

4. Name conditions under which each of the following heating systems might be found eminently fitted:

a. Mechanical warm air.
b. A floor furnace.
c. A one-pipe steam system.
d. Circulating hot water.
e. A heat pump.

5. If you were required to design a heating system for a one-story basementless residence of five rooms and bath, and were limited to an expenditure of $1,000, what system would you adopt? Why?

6. A school building three stories high and heated by circulating hot water, the boiler being fired by a mechanical stoker, plans a new one-story wing for each end. Each of the new wings will contain two classrooms. No more than $1,500 is available for the heating of the new structures. The chimney is approximately in the center of the existing building. How would you heat the new wings? Why?

CHAPTER 20

INCINERATORS AND DESTRUCTORS

The Purchaser's Problems

Basic Considerations. Heating engineers and heating contractors frequently are asked as an integral part of their responsibilities to design or to furnish some form of incinerator or destructor. Incidentally, the significance of the two names for what is fundamentally the same piece of equipment is mainly size. When we talk of capacity in bushels, the chances are the supplier will call his unit an incinerator, but a pounds per hour reference running from perhaps fifty to many thousands of pounds per hour brings out the more imposing name of destructor.

Our first problem then is to determine the capacity of the equipment we need, plus data on the nature of the material to be destroyed. Suppose that we consider each of these factors in order.

TABLE 20.1. AMOUNT OF REFUSE PRODUCED BY VARIOUS ACTIVITIES*

Activity	Weight of refuse, lb	Basis
Hotels	2	Per guest room per day
First-class restaurants	1¼	Per meal
Average restaurants	¾	Per meal
Cafeterias	¾	Per meal
Lunchrooms	⅜	Per meal
Hospitals	8	Per bed per day
Schools	9	Per classroom per day
School lunchrooms	1/12	Per pupil per day
Residences	4	Per person per day, including servants
Apartment houses:		
Kitchenette-type	2	Per day per kitchenette apartment
Larger apartments	7½	Per day per apartment

* Maximum hour $= \dfrac{\text{Daily total, lb}}{24} \times 2.5$.

Based on data provided by Joseph Goder Incinerators.

Estimating the Size Needed. While any appraisal of conditions is certain to vary somewhat with local conditions, the data in Table 20.1 will be helpful.

Nature of the Refuse. Everything that nobody wants, and a good many things that are wanted, eventually end up in incinerators. Usually we can expect to handle not only combustible materials but also bottles, cans, kitchen utensils, silverware, and plain water, as well.

Of these noncombustibles, only the moisture item is of first-rate importance to the designer. It is important because the greater the amount of moisture to be evaporated, the less the net capacity of the machine becomes. Generally, if we are in doubt as to the percentage of moisture to be handled, an allowance of 40 per cent in the total weight will be about right.

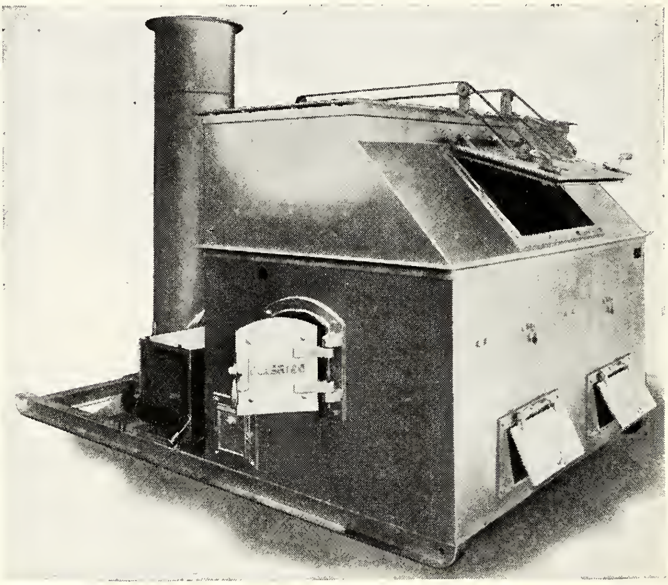


Fig. 20.1. Portable incinerator. Water coils may be installed if desired, producing hot domestic water or steam as a by-product. *(Courtesy of The Plibrico Jointless Firebrick Company.)*

Charging Methods. In apartment houses, a charging chute frequently is built from the incinerator to the roof, the tenants on each floor being provided with a door into this chute.

For the smaller installations, the incinerator must be visited in person, and the refuse dumped through a suitable door in the top, front of side, to suit the convenience of the location. Figure 20.1 shows a so-called "portable unit," with the charging door well placed on the sloping front. The description "portable" really means that the machine is delivered all in one piece, or as a single package.

Spark Catchers. Most communities have ordinances which require the installation of municipally approved spark arresters. Some of these are simply cages formed of screen and mounted on top of the chimney. Others are both elaborate and expensive, involving ducts, fans, and bulky separation chambers or boxes.

Before deciding on any type of incinerator, we should check the local legal requirements for the control of sparks and fly ash. The information so uncovered may quite likely persuade us to give up the idea of our own incinerator and decide to patronize the city plant.

Forms of Incinerators

Procedure of Burning. Since there is a great deal of moisture to be evaporated from garbage and similar refuse, an efficiently designed incinerator usually incorporates some sort of drying shelf, or grate, on which the material is permitted to lie for an appreciable time before actually entering the combustion chamber. To increase the efficiency of the apparatus further, preheated air is not uncommon, the combustion air being forced through hollow walls or ducts where it picks up heat from the furnace.

The presence of moisture and considerable noncombustible material makes ignition difficult at times; consequently some form of auxiliary fuel burner is usually standard equipment. A gas line seems to be preferred where gas is available, but oil and even coal and wood will serve very satisfactorily, if the incinerator is designed for them. The auxiliary fuel may be shut off once energetic combustion is established.

An arrangement of step grates and flat grates satisfies most requirements. The flat grates are of either sliding or tilting design, readily operable from outside the machine, and when opened permit the cans, bottles, and ashes to fall into the collection chamber at the bottom of the machine.

Draft Problems. The small to medium sizes operate on natural draft, but large installations, say those with capacities of more than 1,000 lb of refuse per hour, are likely to require forced draft. The amount of draft necessary will be stipulated by the manufacturer of the equipment, based on some expressed percentage of moisture being present.

When forced draft is employed, there is considerable likelihood of the incinerator being used rather continuously. If this is indeed the situation, it is possible to obtain steam generating boiler sections or coils, by which much of the otherwise wasted heat can be reclaimed. This is a good plan if the economics show it can be justified.

Commercial Sizes. While the larger sizes are all tailor-made installations, we can buy off the shelf a number of small-to moderate-sized units, somewhat in accordance with the data in Table 20.2.

TABLE 20.2. COMMERCIAL SIZES OF COMMON INCINERATORS

Capacity of unit, bu*	Width	Length	Height	Weight, lb	Flue size, in.
1½	1′9″	1′9″	3′10½″	625	7
3	2′1″	†	2′9″	300	7
9	3 0″	3′9″	3′6″	1,000	9
16‡	3′6″	3′11½″	6′6″	3,284	12 × 12
30‡	4′2″	5′9½″	7′6″	7,800	18 × 18
50‡	5′2″	5′9½″	7′6″	11,000	24 × 24

* In converting pounds per hour to bushels, assume that 1 bu of capacity is equal to a burning rate of 8 lb per hr.

† Indicates a round apparatus.

‡ These are built of brick.

Questions

1. How large an incinerator would you need for your own home?
2. If you were the mechanical designer of a six-story apartment house, what type of incinerator, or destructor, would you specify? Why?
3. If the apartment house were 20 stories high, what type of incinerator would you specify?
4. What kind of spark arrester or fly-ash catcher would you install?
5. What charging method would you prefer? Why?
6. What is the purpose of the drying shelf?
7. What is the purpose of auxiliary fuel?
8. Would you attempt to recover any of the heat of combustion? Give reasons for your decision.

CHAPTER 21

SPECIFICATIONS AND THE ANALYSIS OF BIDS

Specifications

Purpose of a Specification. Every specification is intended to accomplish these three purposes:

1. To set forth exactly what the purchaser wants.
2. To mark off the individual responsibilities of the supplier and the purchaser.
3. To provide a common measure, by which competitors for the order may be fairly judged.

Every specification incorporates three major divisions. In the beginning, we present the *general clauses*, covering such nonspecific factors as permits, insurance, bonds, etc. Next, we present clauses that describe in detail the equipment to be furnished and the work to be done, at the same time indicating who does what, when cooperation of the trades is required. For the final division, we reserve our demands for tests, for the guarantee, and for bonus and penalty payments, if these are involved.

How each of these divisions is made will be illustrated subsequently. As for the wording, we may choose between two approaches.

The first way is the older, and may be termed the "shall and will" approach. This technique calls for the use of full and rounded sentences in which the supplier is told what he *shall* do and what other contractors and the purchaser *will* do.

The newer way is called by some of its users the "streamlined" method. Clauses shaped to the streamlined technique are notably innocent of shall and will. The general appearance of the streamlined specification is that of an index, rather than a text, clearcut and to the point.

Practitioners of the streamlined approach are strongly partisan, but this writer, although preferring it, never has been able to conserve on either time or paper by means of the newer approach. This fact is offered simply as the experience of one individual; each of us must choose the method best suited to himself.

The General Clauses. In large projects, the heating equipment is merely a component of the whole; consequently the general clauses set forth at the beginning of the project specification cover the heating items as well as all others. No general clauses for the heating section as such are therefore needed in these cases, save perhaps those that apply peculiarly to the trades involved.

When the heating project is the whole project, then some clauses of a general nature become essential, and we may expect to find them in the first division of the specification substantially as follows:

1. *Scope.* This clause frequently begins: "This specification, in conjunction with drawings numbers ___ ___ ___, is intended to cover the furnishing of all labor and material necessary to provide a complete ___(Type)___ heating system at ___(Location)___, all in accordance with the following details." The purpose of the scope is to show bidders at once the over-all boundaries of the work. This clause is more or less standard even for heating systems that are merely part of a much greater project.

2. *Permits, Licenses, Ordinances, and Laws.* Under this heading we direct the contractor's attention to his responsibility for obtaining all permits necessary to practice his trade, and place on him beforehand any blame that might accrue from the breaking of existing laws.

3. *Insurance and Bonds.* If we ask the contractor to protect the owner with Workmen's Liability, Property Damage, and/or other insurances, and if we demand of him a performance bond, here is the clause in which so to inform him.

4. *Patents.* Under the law, users of a patented device may become jointly liable with the supplier for damages assessed for infringement. If our specification enters pioneering country, we should include under this heading a statement which requires the supplier to save the purchaser harmless from infringement charges.

5. *Signature of Contracts.* Sizable contracts sometimes have the power to break a contractor, if he has slipped seriously in his estimating. Occasionally, we may encounter this situation, with the contractor discovering his predicament early in the construction program, and there may be a tendency to repudiate the contract. In the circumstances, if the contractor states that the individual who signed the contract was without legal power to do so, and is able to substantiate his statement, the purchaser may find himself in the unhappy position of having work started without any assurance of who will finish it. Delays and expense then become inevitable.

In different circumstances, an overenthusiastic salesman may give away more than his company will approve; consequently we should be sure that any signature we accept carries the legal right of commitment, so far as the signer's company is concerned.

The Detail Clauses. In this division, we cover the technical aspects of the material and its installation. As illustrations of the proper approach, here are typical items:

1. *Boiler Particulars.* This clause sets forth all that the bidders must know regarding boiler capacity, working pressure, controls, firing equipment, etc.

2. *Breeching.* Steel thicknesses, supports, treatment for corrosion before erection, are detailed here.

3. *Electrical Work.* Who is responsible for the electrical wiring, designations of the current characteristics, and also equipment standards and limitations appear in this clause.

4. *Insulation.* Here we detail the thickness of insulation and its material for the various units to be covered, such as piping, ducts, breeching, etc. Methods of application also are specified in this section.

Altogether for the detail division of a specification, we may expect to prepare from 10 to 20 clauses. Every single unit of size or importance must be particularized.

The Closing Clauses. With the general and technically detailed clauses completed, we are ready for the closing division of the specification. In this division we include the following:

1. *Starting Up and Tests.* Here we stipulate when and how the new system is to be started up, followed by our requirements for the nature and duration of the acceptance tests.

2. *Guarantee.* General practice customarily makes the installing contractor responsible for his material and workmanship during a period of one year, beginning with the date the system is accepted, in writing.

3. *Visit to the Site.* Bidders customarily are invited to visit the site prior to bidding, in order to see for themselves what conditions of erection, storage, and even delivery of materials are involved. In this clause, the purchaser disclaims any liability for failure by the contractor to evaluate conditions.

4. *Penalties and Bonuses.* Here is the place to declare their presence, magnitudes, and conditions of payment.

5. *Data Submitted with Bids.* We call for information on weights, dimensions, efficiencies, capacities, etc., items on which we will base the comparisons of the various bids.

6. *Submission of Bids.* Generally this is the last clause of the specification. It tells bidders when we want the bids, stipulating both date and hour, and furthermore identifies the address to which the bids must be delivered.

The Prudent Rule. Specifications frequently tend to become lengthy and their preparation a chore. For these reasons we are tempted at times to assume that bidders will include this or that as a matter of course. There is no profit in this assumption, however, for some will, while others will not, and we lose considerable time eventually in obtaining statements and confirmations.

The prudent rule in specification writing is this: If in doubt about an item, put it in.

Contract Documents

Heating engineers and contractors encounter most often the standard contract form issued by the American Institute of Architects.

Many of the clauses in this contract duplicate, except for exact wording, clauses already included in the specification. Others, concerning liens and the right to terminate the contract accorded to both the owner and the contractor, appear in the contract for the first time. A short form of this contract is available if desired, only four pages in length.

The contract documents normally include this AIA contract (or some other contract form), the drawings, and the specifications.

Copies of the contract forms may be obtained from the American Institute of Architects, the Octagon, Washington, D.C.

Analysis of Bids

The important thing to remember when we receive bids in this: The lowest quoted price is not necessarily the cheapest. Items most prominently affecting the total owning cost over a period of years, and which vary with the job, are these:

1. *Efficiency of the boiler.* A cheaper boiler with lower efficiency actually can become more costly in time than a high-priced boiler that operates with high efficiency, due to the extra fuel burned by the former.

2. *Maintenance.* Overstressed materials and overeconomical design eventually plague the temper and pocketbook of the owner.

3. *Electrical costs.* If we attempt to hold down duct or piping costs by cutting the sizes of these fluid conveyors below recommended practice, we eventually pay out the difference in increased power bills for the pumps and fans.

4. *Space requirements and the effect on the building structure.* If floor area is critical, a common situation in many commercial and apartment installations, a large plant may require the erection of a separate building, whereas a smaller one might conceivably be squeezed, although painfully, into some out-of-the-way corner. If floor area is valuable, the space occupied by the plant should be charged against the plant at the prevailing square foot rate.

How the Analysis Is Made. Our first duty, after bids are received, is to examine each for its compliance with the specifications and drawings. Those which fail to meet requirements are rejected.

We are now ready to carry out an economic analysis of the prices and equipment, using data supplied by the bidders. Suppose, as an illustration of the analysis procedure, we have received three bids, identified as *A*, *B*, and *C*, with necessary evaluation data tabulated as follows:

BIDDERS' DATA

Item	Bid *A*	Bid *B*	Bid *C*
Quoted price	$15,425	$13,750	$15,900
Boiler efficiency, per cent	80	75	78
Motor horsepower demands	15	15	12
Foundations, ft	6 × 8 × 1	7 × 10 × 1	6 × 8 × 1
Filters	Cleanable	Throwaway	Throwaway
Total space, ft	12 × 15	16 × 20	14 × 18
Chimney height, ft	70	75	65

The actual investment costs for these competing systems may be developed thusly:

TOTAL INVESTMENT

Item	Bid *A*	Bid *B*	Bid *C*
Quoted price	$15,425.00	$13,750.00	$15,900.00
Foundations @ $75 per cu yd*	133.25	194.50	133.25
Fireproofing over boiler @ $2 per sq ft	96.00	140.00	96.00
Electric service	125.00	125.00	125.00
Chimney cost	500.00	525.00	475.00
Total investment	$16,279.25	$14,734.50	$16,729.25

* Includes excavation, forms, labor, etc.

Now we must calculate the comparative annual operating costs, and here is the way we do it:

ANNUAL OPERATING COSTS

Item	Bid *A*	Bid *B*	Bid *C*
Space @ $2 per sq ft	$360.00	$640.00	$504.00
Interest, 2% of investment	325.59	294.69	334.59
Amortization, based on 20 years	813.96	736.73	836.46
Maintenance (reflecting the cleanable filters in bid *A*)	125.00	160.00	160.00
Electricity	150.00	150.00	120.00
Fuel,* based on 20,000,000 Btu per day, required for 300 days. Oil at 19,000 Btu per lb and 10 cents per gal	4,935.90	5,265.00	5,062.50
Total annual costs	$6,710.45	$7,246.42	$7,017.55

* These fuel charges reflect the differences in boiler efficiency as indicated by the bidders.

The actual owning cost of these systems, compiled from both the initial investment and the annual charges, is calculated as follows:

ACTUAL OWNING COST, BASED ON 20-YEAR LIFE

Item	Bid *A*	Bid *B*	Bid *C*
Total operation	$134,209.00	$144,928.40	$140,351.00
Initial investment	16,279.25	14,734.50	16,729.25
Total owning cost	$150,488.25	$159,662.90	$157,080.25

On the basis of the over-all analysis, therefore, the lowest price quoted actually becomes the most expensive system to own. In this illustration, boiler efficiency and charges for space occupied become the deciding factors. Without an analysis, their importance would be overlooked.

Judgment in the Analysis. The estimate of maintenance always is open to some question. Operating skill affects repair charges seriously, both in frequency and magnitude, consequently when the economic analysis shows bids to be sharply competitive, we should be prepared to defend our conclusions with both facts and sound reasoning.

Some engineers merely assign a percentage of the quoted prices (perhaps 3 or 5 per cent) to the annual charges, and call that the maintenance estimate. This practice is easy but unfair, since the end effect is merely to penalize the higher bidders for their higher prices, and ignores the possibility of these higher prices being the direct result of qualities that actually depress maintenance.

If we have maintenance experiences of our own relative to the equipment under consideration, we have solid grounds on which to base our estimates. In the absence of these experiences, the conscientious engineer, particularly if expensive machinery is involved, will obtain data on breakdowns and repairs from other users. However, information so obtained must be weighed carefully, for the skill of these other users may be greater or less than our own. In any event, whatever we charge against a bid for expected annual maintenance should be the result of thinking, not guessing.

The problem of equipment life and the interest to charge against the capital investment involves personal opinions and the customs of the locality. Generally, 20 years is considered a reasonable life for heating equipment; nevertheless 25 or 30 years is quite probable with our modern designs and intelligent care. As for interest, our devaluated currency and plenitude of investment funds suggest that 2 per cent is a far more realistic charge than the traditional 6 per cent. Low interest charges, of course, decrease the importance of the initial cost while increasing the ultimate effect of the operating charges. Here again we must exercise sound judgment based on existing facts, for our choice of an interest rate whatever it happens to be can become the deciding factor in an economic balance.

The question of space charges seldom occurs where residential installations are concerned. Commercial and industrial heating, however, is another matter. Frequently floor space assigned to the heating plant must be deducted from otherwise productive areas, and in these cases, the plant certainly should bear its share of the cost of doing business.

The annual floor charge assessed against a plant should be based on the average per square foot rate of the floor on which the plant is located. Total charges then are given their comparative weights in the analysis of the bids.

Questions

1. Describe the threefold purpose of a specification.
2. What are the three major divisions of a specification?
3. Outline a specification for a one-pipe steam system.
4. Outline a specification for a forced circulation warm air system.
5. Outline a specification for a forced circulation hot water system.
6. What is the prudential rule?
7. Name the contract documents.
8. What are the four items that affect most strongly an analysis of bids?

9. With the bidders' data at hand, describe the three steps that discover the actual owning costs.

10. Why is an analysis important?

11. For practice, tabulate three theoretical quotations, and analyze them.

12. How would you proceed on the estimating of maintenance?

INDEX

A

Absolute humidity, definition of, 12
Absolute viscosity, definition of, 21
  of fuel oils, 321–323
Absolute zero, definition of, 1, 276
Absorbent, definition of, 1
Activated carbon in filters, 250–252
Adiabatic, definition of, 1–2
Adsorbent, definition of, 1
Aerosol, definition of, 2
AGA ratings of gas-fired boilers, 82–83, 85
Air, in combustion, of artificial gas, 347–349
    of bagasse, 351–352
    chemistry of, 352
    of coal, 341–343
    of liquefied petroleum gases, 351
    moisture in, 359–360
    of natural gas, 349–350
    of oil, 346–347
    table of, 354
    of wood, 350–351
  free, definition of, 9–10
  in hot water heating systems, 121–122
  as source of heat, 278–282
  in steam heating systems, 38
  velocity of, 259
  volume of, 30
    at various altitudes, 95
  warm (*see* Warm air heating systems)
  weight of, 30
Air conditioning, with heat pumps, 293, 295–296
  and warm air heating, 180
Air evaporators, inside, 281
  limitations of, 278, 281–282
Air filters (*see* Filters)
Air-horsepower equation, 243, 245
Air jets in smoke prevention, 364
Air movement, checking of, 221–222
Air valve in steam heating systems, 42
Air venting in hot water heating systems, 121–122
Altitude, and air volume, 95
  and boiler capacity, 93–95
Aluminum insulation, 402
American Gas Association (AGA), ratings of gas-fired boilers, 82–83, 85
  rules for gas flues, 389–391
American Petroleum Institute (API) scale for fuel oils, 311, 313
American Society of Heating and Ventilating Engineers (ASHVE) tests of steam in radiators, 37
American Society for Testing Materials, Standard Viscosity-Temperature Chart, 324
  values of pyrometric cones, 408
Anthracite coal, 301
  specifications for, 298
Armour Research Foundation, Illinois Institute of Technology, 277
Artificial gases (*see* Gases, artificial)
Asbestos air-cell insulation, 399–400
Asbestos millboard, 400–401
Asbestos paper, 400
Asbestos sponge felt, 398
Ash collectors, 137–138, 141
Ashes, carbon in, 360
  coal, handling of, 306–308
Aspect ratio, definition of, 2
Asphalt roofing, 406
Atmospheric steam heating systems, 54–55
Atomic weights, table of, 353
Automatic controls in hot water heating systems, 118–120

B

Bagasse, analysis and combustion of, 351
Barometric dampers, 92
Baseboard heating systems, 237–238
  characteristics of, 427–428
Baumé hydrometer scale, 313
Beams, horizontal, piping around, 57
Bel, definition of, 6
Bids for heating systems, 438–441
  data on, 439–440
Bituminous coal, 300–301
  analysis of, 343
Blow, definition of, 2
Blowers, 244–246

Boiler firing equipment, coal firing apparatus, 127–141
  gas burners, 151–154
  oil burners, 141–151
Boiler fittings, 91–92
Boiler horsepower, definition of, 2, 5
Boiler insulation, 92–93
Boiler test, 360–362
Boilers, capacity of, and altitude, 93–95
  cast-iron, 61–66
    forms of, 64–66
    ratings of, 61–64
  combustion test of, 360–362
  electric, 90–91
  gas-fired, 79–85
    ratings of, 82–83, 85
  general description of, 61
  heat losses in, 356–360
  in hot water heating systems, sizing of, 124–125
  ratings of, AGA, 82–83, 85
    and chimney draft, 372
    IBR, 61–64
    SBI, 67–68
  and steam generators, 85–88
  steel, 67–68
    downdraft furnace, 75–76
    firebox, 72–77
    magazine, 78, 80
    return tubular, 71–73
    round, 68–70
    Scotch, 70–71
    smokeless arch, 75
    vertical, 76–78
  waste-heat, 88–90
Breechings of chimneys, design of, 370–371, 374–375
  insulation of, 405–406
Brick chimneys, 376–378, 382–383
Btu, conversion of, to cfm, 195–196
  definition of, 5
Bucket-a-day water heaters, 156
Butane, 330, 351

C

Cannel coal, 301
Carbon, activated, in filters, 250–252
  in ashes, 360
  in combustion, 341, 352, 354
Carbon dioxide in combustion, 360–362
Carbon monoxide in combustion, 360–362
Cast-iron boilers (*see* Boilers, cast-iron)
Ceilings, coefficients of heat transmission through, 29
Centigrade scale, conversion to Fahrenheit, 20
Centipoise, definition of, 22
Centistoke, definition of, 21, 321
Cgs system, explanation of, 22–23
Chimney flues, design of, 373
  for fireplaces, 414–415
  gas (*see* Gas flues)
  linings of, 373, 394–395, 415
  sizes of, 374
Chimney pots, 386–387
Chimneys, breechings, 370–371, 374–375
    insulation of, 405–406
  brick, 376–378
    radial, 382–383
  cleaning of, 375
  commercial and industrial, 376–386
  concrete, 383–385
  covers on, 386–387
  definition of, 367
  design of, 367–372
    for safety, 373–374
  downdraft in, 392
  draft in, 367–368, 371–372
    faulty, 391–392
  friction losses in, 368–372
  for gas-fired furnaces, 387–391
    American Gas Association rules for, 389–391
    National Board of Fire Underwriters rules for, 388
  lightning protection for, 392–394
  lightweight, 375–376
  rules of practice for, on breechings, 374–375
    from experience, 373–374
    Underwriters', 372–375
  safety in, 372–375
  smoke prevention in, 392
  steel, anchor bolts for, 378
    foundations for, 379–381
    with guys, 381–382
    painting of, 382
    self-supporting, 378–381
  venturi stacks, 384–386
  for warm air heating systems, 197
  wind pressure on, 377

Cinder catchers, 137–138
  sizing of, 138–141
Circulation in hot water heating systems, forced, 107–109
  natural or gravity, 97–99
Circulators for fireplaces, 418–421
CO (*see* Carbon monoxide)
CO_2 (*see* Carbon dioxide)
Coal, analysis of, 342–346
  anthracite, 301
    specifications for, 298
  bins and bunkers for, 303–304
    sizing of, 305–306
  bituminous, 300–301, 343
  cannel, 301
  characteristics of, 298–302
  combustion of, 341–343
  pulverized, 135–138
  specifications for, 298, 299
  storage of, 302–306
Coal ashes, handling of, 306–308
Coal firing apparatus, 127–141
  (*See also* Stokers)
Coefficients of heat transmission, application of, 32–33
  calculation of, 26–28
  through ceilings and floors, 29
  through glass and doors, 29
  through walls, 28
Coke, characteristics of, 302
Cold springing in piping, 122
Combustion, air requirements for various materials, 354
  of artificial gas, 347–349
  of bagasse, 351–352
  boiler test of, 360–361
  of carbon, 341, 352, 354, 360
  carbon dioxide and carbon monoxide in, 360–362
  chemistry of, 352, 353–355
  of coal, 341–343
    proximate analysis of, 343–346
  efficiency determinations in, 356–359
  emissivity in, 359
  of fuel oils, 346–347
  gases as products of, 355, 387
  heat balance in, 360
  heat losses in, 356–360
    from moisture in air, 359–360
  of hydrogen, 342, 345, 354, 357
  of liquefied petroleum gases, 351
Combustion, of natural gas, 349–350
  principles of, 341
  products of, 355–356
  smoke in, 362–366
  of sulfur, 342, 354–355
  of wood, 350–351
Compression tanks in hot water heating systems, 117
Concrete chimneys, 383–385
Conductance, definition of, 5
  of various materials, table of, 27
Conductivity, definition of, 5
Controls, automatic, in hot water heating systems, 118–120
Convection, definition of, 5
Convection losses in combustion, 357–359
Convectors, definition of, 6
Conversion burners, gas, 153–154
Converters in hot water heating systems, 118–119
Cooling systems, radiant, 238–239
Copper pipe, dimensions of, 123
Corkboard insulation, 401–402
Corrosion in water heaters, 178

D

Dampers, barometric, 92
  in gravity furnaces, 194–195
Decibel, definition of, 6–7
Degree-days, definition of, 7
  table of, 8
Density, definition of, 8–9
Design temperatures, 24–25
Destructors (*see* Incinerators)
Desuperheating of steam, 269
Dew point, definition of, 7–8
Dichlorodifluoromethane, properties of, 279–280
District steam systems, construction, cost of, 265
    underground, 266–267
  control of, 274
  flowmeters in, 261–263
  hot water from, 174–175
  manholes in, 269–272
  overhead mains in, 272
  pressure in, 266–268
  rates of, consumption metering, 260–263
    demand metering, 264–265
    space or radiation charge, 260

District steam systems, return of condensate in, 266
  service connections in, 269–273
  temperature in, 269
Domestic water heaters (*see* Water heaters)
Doors, coefficients of heat transmission through, 29
Downdraft furnaces, 75–76
Draft, in chimneys, 367–368, 371–372, 391–392
  in fireplaces, 417
Drainage in steam heating system, 37
Ducts (*see* Warm air heating systems, ducts)
Dynamic head, definition of, 11
Dyne, definition of, 21

E

Earth, as source of heat, 277, 287–290
  temperature of, 26
Electric air heater, 256–257
Electric eye in oil burners, 148–149
Electric panel heating, 235–237, 256–257
Electric water heaters, 164
Emissivity in heat losses, 359
Enthalpy, definition of, 9
Entry head, definition of, 11
Equivalent direct radiation, definition of, 9
Equivalents, definition of, 9
  table of, 10
    metric system, 23
Evasé stacks, 385–386
Expansion of piping, 122
Expansion tanks in hot water heating systems, 115–116
Explosions in water heaters, 156, 160
Exposure allowance in design, 26

F

F-12, properties of, 279–280
Factories, water heaters in, 168–171
Fahrenheit scale, conversion to centigrade, 20
Fans, centrifugal, 245–246
  characteristics of, 243
  for corrosive service, 247–248
  horsepower of, 243, 245
Fans, performance curve of, 244
  propeller, 246–247
  size of, 247
  tubeaxial, 246
  types of, 245–246
  vaneaxial, 246
Ferrous materials, description of, 62
FHA requirements for residential structures, 25, 33–34
Filling densities for liquefied petroleum, 331–332
Filters, activated carbon in, 250–252
  selection of, 252
  types of, 248–250
Fire clay, 406–407
  pyrometric cone equivalents of, 408
Firebox boilers, 72–77
Firebrick, pyrometric cone equivalents of, 408
  standard dimensions of, 409
Fireplace circulators, 418–421
Fireplace screens, 421
Fireplaces, circulating, 418–421
  design of, 412–416
    new British, 415–416
  dimensions of, 413
  draft in, 417
  flues for, 414–415
  open-hearth, 417–418
  smoky, 417
  starting fires in, 416
Flame rod in oil-fired furnaces, 148
Floor furnaces, 255–256, 428
Floors, coefficients of heat transmission through, 29
  steam piping in, 58
Flowmeters in district steam systems, 261–263
Flues (*see* Chimney flues; Gas flues)
Fly-ash collectors, 137–148, 141
Forced circulation in hot water heating systems, 107–109
Foundations for chimneys, 379–381
Free air, definition of, 9–10
Freon, definition of, 10–11
Friction head, definition of, 11
Friction losses, in chimneys, 368–372
  in fuel-oil piping, 325–326
  in steam heating systems, 48
  in warm air heating systems, 205, 208–209, 217–218

Fuel-oil piping, equivalent pipe lengths, 326
  friction loss in, 325–326
  materials for, 319–320
  pump discharge, 327
  sizing of, 321–323
    nomographs for, 327–328
  suction, 323–326
Fuel-oil tanks, 314–317
Fuel oils, analysis of, 346
  Baumé-API scale for, 313
  color of, 312
  combustion of, 346–347, 355
  distillation of, 312
  flash and fire points of, 311–312
  gravity of, 311, 313
  pour test of, 312
  pumps for, 317–319
  specific gravity of, 321–323
  specifications for, 310–311
  units of volume of, 313
  viscosity of, 312, 321–324
Furnace coils as water heaters, 164–165
Furnaces, design of, 410–411
  downdraft, 75–76
  floor, 255–256, 428
  gas-fired, characteristics of, 428
    combustion in, 387
    draft hoods on, 387–388
    vents and flues for, 387–391
      American Gas Association rules on, 389–391
      National Board of Fire Underwriters rules on, 388
  gas-oil, dimensions of, 410
  gravity, 192–197
    balancing of, 195
    chimneys for, 197
    dampers in, 194–195
    ratings of, 194–195
  oil, dimensions of, 410
  smoke prevention in, 364–365
Fuses, sizes of, for electric water heaters, 164

G

Gas burners, applications of, 152–158
  conversion, 153–154
  principles of, 151–152
  sizing of, 153–154
Gas-electric ignition, 148
Gas-fired boilers, 79–85
Gas-fired furnaces (*see* Furnaces, gas-fired)
Gas-fired water heaters, 157–158
Gas flues, American Gas Association rules for, 389–391
  capacities of, 391
  National Board of Fire Underwriters rules for, 388
  sizing of, 388–389
Gas lines, piping in, 331, 337–339
  sizing of, 333–336
    charts for, 335–336
    formulas for, 333–334
Gas tanks, cylinder systems, 332–333
  filling densities for, 331–332
Gases, artificial, 329–330
    analysis of, 347
    combustion of, 347–349
  as combustion products, 355, 387
  handling of, 330–332
  liquefied petroleum, 330–331, 351
  natural, 328–329
    analysis of, 349
    combustion of, 349–350
  safety rules for, 339
  types of, 328
Geepound, definition of, 22
General Electric psychrometric chart, facing p. 8
Generators, steam, 85–88
Glass, coefficients of heat transmission through, 29
Glass insulation, 401
Grates, handfired, 68, 127
Gravity circulation in hot water heating systems, 97–99
Gravity furnaces (*see* Furnaces, gravity)
Gravity systems in warm air heating, 181–182
Ground coils, heat from, 277, 287–290
Ground temperatures, 26

H

Hartford connection in steam heating systems, 50
Head, definition of, 11
Heat, latent, 13, 37
  loss of, in combustion, 356–360

Heat, solar, absorption of, 161–162
  specific, definition of, 18–19
Heat insulation (*see* Insulation)
Heat pumps, air evaporators, 278, 281–282
  capacity of, 291, 294, 296
  characteristics of, 428
  combination designs, 290–292
  for cooling in summer, 293, 295–296
  cost of, 285–286, 293
  ground coils, 277, 287–290
    advantages of, 290
    principles of, 287–289
    sizing of, 289–290
  principle of, 276–278
  size of, 294, 296
  and storage tanks, 292
  water, 277, 282–287
    advantages of, 287
    calculations for, 284–286
    cost of pumping and compressing, 285–286
    disposal of water from, 286–287
    temperature in, 282–283
Heat transmission (*see* Coefficients of heat transmission)
Heatilator for fireplaces, 419–420
Heating, radiant (*see* Radiant heating)
Heating loads, calculation of, 24–31
    typical procedures in, 31–33
Heating panels (*see* Panel heating systems)
Heating systems, bids for, 438–441
  choice of, 422–425
  comparison of, 425–428
  contracts for, 436, 438
  cost of, 422–423
  location of, 429
  special requirements for, 423–425
  specifications for, 435–438
  (*See also* Hot water heating systems; Panel heating systems; Steam heating systems; etc.)
Heights, relative, and circulation, 98–99
High buildings, hot water heating systems in, 117–119
Horsepower, boiler, definition of, 2, 5
  of fans, 243, 245
Hot water heating systems, air venting in, 121–122
  automatic controls, 118–120
  boilers, 124–125
Hot water heating systems, characteristics of, 425–427
  circulation in, forced, 107–109
    natural or gravity, 97–99
  closed tanks, 115–116
  compression tanks, 117
  converting from gravity to forced circulation, 115
  diversion fittings, 113–114
  expansion tanks, 116
  forced circulation, 107–115
    one-pipe, 112–113
    sizing of piping in, 109–112
    two-pipe, 109–112
  gravity, one-pipe, 105–106
    sizing of piping in, 99–104
    two-pipe, 106–107
  multiple circuits, 117–118
  open tanks, 115–116
  piping in, 122–123
  pumps for, 110–111
  temperature drop in, 121
  water velocities in, 99–100
  zoning, 117–118
Hot water requirements, from district steam systems, 174–175
  in factories, 168–171
  in office buildings and hotels, 172–174
  in residences, 167–169
  in restaurants, 172–173
  table of, 169
  (*See also* Water heaters)
Houses (*see* Residences)
Humidity, definition of, 12
Hydrogen in combustion, 342, 345, 354, 357
Hydrometer, Baumé scale on, 313

I

IBR ratings of cast-iron boilers, 61–64
Ignition in oil burners, 147–149
Ignition arch in coal stokers, 127, 129
Inches, of mercury, definition of, 12
  of water, definition of, 12
Incinerators, burning procedures in, 433
  capacity of, 431–432
  charging of, 432–433
  draft in, 433–434
  sizes of, 434
  spark catchers in, 433

Indoor-outdoor controls in hot water heating systems, 120
Infiltration allowance, 28, 30, 31
Institute of Boiler and Radiator Manufacturers (IBR) ratings of cast-iron boilers, 61–64
Insulating cements, 405
Insulating plaster, 404–405
Insulation, aluminum, 402
  asbestos air-cell, 399–400
  asbestos millboard, 400–401
  asbestos paper, 400
  asbestos sponge felt, 398
  asphalt roofing, 406
  on breechings, 405–406
  corkboard, 401–402
  glass, 401
  good qualities of, 396–397
  high-temperature combinations, 403–404
  on irregular surfaces, 405
  magnesia, 85%, 398–399
  rock wool, 397
  thickness of, 404
  vermiculite concrete, 401
  wood-cellulose, 402–403
Iron, types of, 62
Isothermal, definition of, 12

K

Kent formulas for foundation diameters, 379
Kerosene, properties of, 313
Kinematic viscosity, definition of, 21–22
  of fuel oils, 321–323, 325
Kinematics, definition of, 321
Kinetic Chemicals Company, refrigerants made by, 10

L

Latent heat, definition of, 13
  in steam heating systems, 37
Lift fittings in steam heating systems, 53
Lighting, of fireplaces, 416
  of gas burners, 151–152
  of oil burners, 147–148
Lightning, protection of chimneys from, 392–394
Lignite, characteristics of, 299–300
Liquefied petroleum gases, 330–331, 351
Logarithmic mean temperature difference, 13

M

Magazine boilers, 78, 80
Magnesia, 85%, 398–399
Magnesium rods in water heaters, 178
Manholes, in district steam systems, 269–272
  in oil tanks, 317
Materials, conductances of, table of, 27
  ferrous, description of, 62
Mercury, inches of, 12
Metric system, table of equivalents, 23
Mol, definition of, 14–15
Molecular weights, table of, 14
Monoflo fittings in hot water heating systems, 113

N

National Association of Fan Manufacturers, 243
National Board of Fire Underwriters rules, on chimneys, 372–375
  on gas flues, 388
Natural circulation in hot water heating systems, 97–99
Natural draft in chimneys, 367–368
Natural gas (*see* Gases, natural)
Noise, measurement of, 6–7

O

Odorizing of liquefied petroleum gases, 330
Odors, control of, by filters, 250–252
Oil burners, air atomizing, 142
  electric eye in, 148–149
  furnace dimensions for, 410
  ignition in, 147–149
  mechanical pressure atomizing, 142–144
  preheaters, 145–146
  principle of, 141
  rotary-cup, 145
  safety devices in, 147–149
  sizing of, 146–147
  steam-jet atomizing, 141–142
  vaporizing, 149–151

Oil-fired water heaters, 158–160
Oil piping (*see* Fuel-oil piping)
Oil preheaters, 145–146, 320
Oil pumps, 317–319
Oil tanks, 314–317
Oils, fuel (*see* Fuel oils)
Oliphant formula for sizing gas lines, 334
Open-hearth fireplaces, 417–418
Orifices in steam piping, 58–59
Orsat gas analyzer, 360–361

P

Package steam generators, 86–88
Panel heating systems, characteristics of, 427
  control of, 231–234
  design of, 224–235
  electric, 235–237, 256
  heating agents for, 228
  principles of, 223–224
  radiant glass panels, 237
  temperature in, 224–228
  venting, 231–234
  warm air, 234–235
  water coils, sizing of, 228–231
Peat, characteristics of, 299
Petroleum gases, liquefied, 330–331, 351
Pickup load in hot water heating systems, 125
Pipe fittings in hot water heating systems, friction in, 100–101
Pipes, copper, dimensions of, 123
  fuel-oil, 326–327
  iron, dimensions of, 104
  in warm air heating systems, standard sizes for, 213–214
Piping, fuel-oil (*see* Fuel-oil piping)
  for gases, 331, 337–339
  in hot water heating systems, choice of, 122–123
    expansion and contraction of, 122
    sizing of, 99–104, 109–112
  in steam heating systems, 43–50, 57–58
    orifices in, 58–59
    sizing of, 43–47, 49–50, 52, 54
  in water heaters, 178–179
Poise, definition of, 21
Poisseule equation, 323
Pole formula for sizing gas lines, 333–334
Poundal, definition of, 22
Preheaters, oil, 145–146, 320
Pressure drop, in district steam systems, 267–268
  in steam heating systems, 47–49
Propane, 330, 351
Psychrometric chart, General Electric, facing p. 8
Pulverized coal, 135–138
Pumps, for fuel oils, 317–319
  heat (*see* Heat pumps)
  for hot water heating systems, 110–111, 115
  vacuum, in steam heating systems, 35–36, 52–53
Pyrometric cones, 407–408

R

Radiant cooling, 238–239
Radiant heating, general principles of, 16–18
Radiant panels (*see* Panel heating systems)
Radiation, equivalent direct, definition of, 9
Radiation losses in combustion, 357–359
Radiators, heat emission from, 63–64
  in hot water heating systems, 112, 114
  in steam heating systems, 46–47, 58
  superheated steam in, 37–38
Réaumur scale, conversion to Fahrenheit, 20
Refractories, castable, 409–410
  definition of, 406
  firebrick (*see* Firebrick)
  good qualities of, 407
  manufacture of, 406–407
  pyrometric cones for testing, 407–408
  ramming mixes, 409–410
  specifications for, 407–409
Refrigerants, in heat pumps, 276–277. 280–281
  made by Kinetic Chemicals Company, 10
Refrigeration, reversed, 278
  ton of, definition of, 20
Refrigerators compared with heat pumps, 276
Refuse, amount of, produced by various activities, 431
  nature of, 432

Refuse disposal (*see* Incinerators)
Registers in warm air heating systems, 188–192, 209–212, 221
Residences, calculation of heating loads in, 31–33
  FHA requirements for, 25, 33–34
  infiltration allowance in, 28, 30, 31
  solar heating in, 34
  space heaters for, 254–256
  unheated spaces in, 25–26
  water heaters in, 167–169
  (*See also* various types of heating systems)
Retarders, in boilers, 69
  in gas-fired heaters, 157
Return traps in steam heating systems, 56–57
Return tubular steel boilers, 71–73
Reynolds number, definition of, 15–16
  of fuel oils, 322–323, 325
Ringelmann charts of smoke, 363
Riser circuits in hot water heating systems, 98
Rock wool, 397

S

St. Louis, Mo., ground coils used in, 277, 288
Sawdust as fuel, 309–310
Saybolt viscosimeter, 21, 312, 321
SBI ratings of steel boilers, 67–68
Scotch boilers, 70–71
Segar pyrometric cones, 407–408
Sizing, of boilers in hot water heating systems, 124–125
  of fuel-oil piping, 320–323, 327–328
  of gas burners, 153–154
  of gas flues, 388–389
  of gas lines, 333–339
  of oil burners, 146–147
  in panel heating systems, 228–231
  of piping, in hot water heating systems, 99–104, 109–112
    valve-area method, 103–104
   in steam heating systems, 43–47, 49–50, 52, 54
  of stokers and cinder catchers, 138–141
  of tanks in hot water heating system, 116–117
Sizing, in warm air heating systems, 182–192, 201–203
  of water heaters, 167–174
Skyscrapers, hot water heating systems in, 117–119
Slug, definition of, 22
Smoke, causes of, 363
  definition of, 362–363
  density of, 363
  in fireplaces, 417
Smoke maker for checking warm air systems, 220–221
Smoke prevention, by burning volatiles, 363–364
  in chimneys, 392
  by furnace design, 364–365
  human element in, 365–366
Smokeless arch, 75
Solar heat, absorption of, 161–162
Solar heating, 34
Solar water heaters, 161–164
Soldered joints in copper tubing, 123
Sound, measurement of, 6–7
South Bend, Ind., ground coils used in, 288
Space heaters, characteristics of, 253–254, 428
  direct-fired, industrial, 257–259
    residential, 255
  electric, 256–257
  floor furnaces, 255–256
Specific gravity of fuel oils, 321–323
Specific heat, definition of, 18
  table of, 19
Specifications for heating systems, clauses in, 436–438
  purpose of, 435–436
Spitzglass formula for sizing gas lines, 333
Stacks (*see* Chimneys)
Static head, definition of, 11
Steam, characteristics of, 18–20, 35–37
  dry saturated, definition of, 18
    pressure table, 3
  superheated, action of, 37–38
    definition of, 19
    in district steam systems, 269
    pressure table, 4
Steam generators, 85–88
Steam heating systems, atmospheric, 54–55
  characteristics of, 427

Steam heating systems, design of, rules for, 38–39
    tricks in, 57–60
  with extra drips, 55–56
  hybrids, 54–57
  one-pipe, 39–52
    design of, 39–42
    downfeed, 51–52
    Hartford connection, 50
    pressure drop, 47–49
    return piping, 43, 49–50
    runouts and risers, 42–43
    sizing of, 43–47, 49–50, 52
  return traps in, 56–57
  two-pipe, 52–53
  vapor, 56
  water level in, 41–42
Steam pressure in district steam systems, 266–268
Steam systems, district (*see* District steam systems)
Steel, types of, 62
Steel Boiler Institute (SBI) ratings of steel boilers, 67–68
Steel boilers (*see* Boilers, steel)
Steel chimneys, 378–382
Stefan-Boltzmann equation, 17, 357, 359
Steubenville, Ohio, heat pump used in, 292
Stoke, definition of, 21, 321
Stokers, overfeed, 130
  pulverized-fuel systems, 135–137
  sizing of, 138–141
  spreader, 132–135
  Stowe, 129
  traveling-grate, 127–129
  underfeed, 130–132
Suction piping for fuel oils, 323–326
Sulphur, in combustion, 342, 354–355
  in fuel oils, 312
Sun heat (*see* Solar heat)
Superheated steam (*see* Steam, superheated)

T

Tall buildings, hot water heating systems in, 117–119
Tanks, fuel-oil, 314–317
    embankments and dikes for, 315
    manholes in, 317
    pipe connections in, 316–317
    vents for, 317
  gas, 331–339
  in hot water heating systems, 115–117
Temperature difference, logarithmic mean, 13
Temperature drop in hot water heating systems, 121
Temperature scales, comparison of, 20
Temperatures, design, inside, 24–25
    and low temperatures, 25
    outside, 24
Throw of air, definition of, 2, 20
Ton of refrigeration, definition of, 20
Total discharge head, definition of, 11

U

Underwriters loop, 50
Underwriters' rules, on chimneys, 372–375
  on gas flues, 388
Unheated spaces, temperature of, 25–26
Unit heaters, 239–241

V

Vacuum pumps in steam heating systems, 35–36, 52–53
Vapor systems in steam heating, 56
Velocity head, definition of, 11
Velometer, use of, 195–196
Venting in radiators, 121–122
Vents for gas-fired furnaces, 391
Venturi stacks, 384–386
Vermiculite concrete insulation, 401
Vertical boilers, 76–78
Violet-ray lamp, 215
Viscosimeter, Saybolt, 21, 312, 321
Viscosity, of fuel oils, 312, 321–324
  principles of, 20–22
  terms in, 21–23
  (*See also* Absolute viscosity; Kinematic viscosity)
Volatiles in smoke prevention, 363–364
Volcano, waste-heat boiler in, 89–90

W

Walls, coefficients of heat transmission through, 28
  steam piping in, 58

Warm air heating systems, characteristics of, 425
  chimneys for, 197
  ducts, aluminum, 207–208
    friction in, 205, 208–209
    galvanized iron, 205–207
    insulation of, 196–197, 214
    materials and design of, 203–204
    return, 183–186
    sizing of, 201–203
    standard sizes for, 213–214
  friction losses in, 205, 208–209, 217–218
  gravity, design of, 181–182
  gravity furnaces, 192–197
  leaders, insulation of, 196–197
    sizing of, 182–183, 187
  mechanical circulation, 197–218
    air wall, 219–220
    arrangement of, 214–217
    design of, 197–201, 218
    return arrangements for, 212–213
    room induction, 218–219
  pipes and fittings, standard sizes for, 213–214
  principles of, 180–181
  registers, 188–192, 209–212
  return ducts, 183–186
  smoke maker for checking, 220–221
  stacks, sizing of, 182–183, 187–188, 202–203
Warm air panel heating, 234–235
Waste-heat boilers, 88–90
Water, boiling point of, 35
  chemistry of, 354
  in heat pumps, 277, 282–287
  hot (*see* Hot water requirements)
  inches of, 12
Water gauges, pressures in, 12
Water hammer in steam heating systems, 59–60
Water heaters, auxiliary furnace coils, 164–165
  bucket-a-day, 156
  choice of, 165–166
  combination systems, 176–178
Water heaters, corrosion in, 178
  direct-fired, 156–160
    nonintegral, 160
  and district steam systems, 174–175
  electric, 164
  explosions in, 156, 160
  gas-fired, 157–158
  magnesium rods in, 178
  nonfired, 160–165
  oil-fired, 158–160
  open-pipe, 165
  owning cost of, 166
  piping in, 178–179
  sizing of, in factories, 168–171
    in office buildings and hotels, 172–174
    in residences, 167–168
    in restaurants, 172–173
  solar, 161–164
  tankless, 160–161
  temperature in, 176
Water level in steam heating systems, 41–42
Water temperature, control of, in hot water heating systems, 120
Water velocities in hot water heating systems, 99–100, 114–115
Weymouth formula for sizing gas lines, 334
Wind pressure on chimneys, 377
Wiring, sizes of, for electric water heaters, 164
Wood, analysis of, 350
  combustion of, 350–351
  in fireplaces, 416
  as fuel, 308–310
Wood-cellulose insulation, 402–403

Z

Zero, absolute, definition of, 1, 276
Zoning of buildings in hot water heating system, 117–118
Zurich, Switzerland, heated by river water, 277

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Date Due

Due	Returned	Due	Returned
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NOV 28 '60	NOV 17 '60		
MAY 11 '66	MAY 16 '66		
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OCT 19 '67	OCT 16 '67		
OCT 30 '67	OCT 30 '67		
NOV 13 '67	NOV 13 '67		
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